Remarks

Applicant has fully reviewed the Office Action dated September 15, 2011, and offers the foregoing amendments and the following remarks in response.

Each of claims 1-17 stands rejected under 35 USC §112, 1st paragraph as allegedly failing to comply with the enablement requirement. The Federal Circuit has stated that "the test of enablement is whether one reasonably skilled in the art could make or use the invention from the disclosures in the patent **coupled with information known in the art** without undue experimentation." *United States v. Telectronics, Inc.*, 857 F.2d 778, 785, 8 USPQ2d 1217, 1223 (Fed. Cir. 1988). Furthermore, MPEP §2164.01 states that "**a patent need not teach, and preferably omits, what is well known in the art**. *Citing In re Buchner*, 929 F.2d 660, 661, 18 USPQ2d 1331, 1332 (Fed. Cir. 1991); *Hybritech, Inc. v. Monoclonal Antibodies, Inc.*, 802 F.2d 1367, 1384, 231 USPQ 81, 94 (Fed. Cir. 1986), *cert. denied*, 480 U.S. 947 (1987); and *Lindemann Maschinenfabrik GMBH v. American Hoist & Derrick Co.*, 730 F.2d 1452, 1463, 221 USPQ 481, 489 (Fed. Cir. 1984).

With regard to claim 1, line 3, the Action alleges that the specification does not adequately describe what a star train is. Similarly with regard to claim 6, lines 1-2, the Action alleges an inadequate description of a solar train. Initially, Applicant submits that both of these rejections fail to establish a *prima facie* case for lack of enablement. That is, the Action does not allege that *the ordinarily skilled artisan could not make or use the invention* based on the disclosures in the patent coupled with information known in the art without undue experimentation. Rather, the Action simply states that the specification does not adequately describe the respective gear trains. For at least this reason, the rejections should be withdrawn.

Applicant submits that both a star train and a solar train are well known in the art of gear assemblies. Thus, merely naming the gears as a "star train" or "solar train" is sufficient to satisfy the enablement requirement. That is, because an ordinarily skilled artisan understands exactly what a star train is and what a solar train is, the specification need not include a disclosure of the detailed assembly of either type of gear. Specifically, a star train and a solar train are two types of epicyclical gears, the remaining epicyclical gear assemblies including at least planetary trains. Furthermore, these epicyclical gear configurations were well known in the art at the time of filing of the instant application. For examples of disclosures of the specific gear arrangements, see, for example the attached patents as ***Exhibits A-C*** (***Exhibit A***: U.S. Patent No. 4,774,855 to

Murrell, et al., Figs. 4-6, and col. 4, line 38 – col. 5, line 32; ***Exhibit B***: U.S. Patent No. 4,104,932 to Hansson, col. 1, ll. 5-13; ***Exhibit C***: U.S. Patent No. 6,223,616 to Sheridan, col. 1, ll. 14-40). Given the MPEP and Federal Circuit directive *not* to include such information which is well known in the art, the instant specification does not include detailed description of the various types of epicyclical gears. In further view of this well-known nature of the various epicyclical gear configurations, any reference to a star train, a solar train, or a planetary train fully satisfies the enablement requirement of §112, 1st paragraph. Accordingly, Applicant respectfully requests favorable reconsideration.

Additionally, claims 1-17 stand rejected under 35 USC §112, 2nd paragraph as allegedly indefinite for failing to particularly point out and distinctly claim the subject matter which applicant regards as the invention. According to MPEP §2173.02,

> Definiteness of claim language must be analyzed, not in a vacuum, but in light of: (A) The content of the particular application disclosure; (B) **The teachings of the prior art**; and (C) **The claim interpretation that would be given by one possessing the ordinary level of skill in the pertinent art** at the time the invention was made.

Here, the terms "high ratio," "high torque," and "low torque" are objected to as allegedly indefinite with no standard for ascertaining the requisite degree in the claims. Without any comment on the adequacy of the rejections, Applicant has amended all claims to remove reference to these allegedly objectionable phrases. Furthermore, claim 1 has also been amended as suggested in the Action to remove or amend the allegedly objectionable phrases "or other," "the combined power," and "the total torque."

The Action further alleges that claims 1 and 6 fail to comply with §112, 2nd paragraph as being indefinite for the alleged lack of an adequate description of a star train and a solar train. In this rejection, no consideration is given to the teachings of the prior art or the claim interpretation that the ordinarily skilled artisan would give the claims. For the reasons stated above, both a star train and a solar train are well known gear configurations, with adequate description in the teachings of the art (see, e.g. attached ***Exhibits A-C***). In light of at least these references, it becomes clear that the claim language of claims 1 and 6 with respect to star trains and solar trains is not indefinite.

Similarly with regard to claim 7, the Action alleges a failure to further limit claim 6 from which claim 7 depends. Specifically, it is alleged that a solar train appears to be a planetary

train. Applicant notes that a *rejection* of a dependent claim for failing to further limit the subject matter of a previous claim is improper (see MPEP §608.01(n)II). If such a situation were the case, the claim should be objected to, rather than rejected. Regardless, this allegation is not true with respect to claim 7. The art teaches, and an ordinarily skilled artisan understands that a planetary train is a *different epicyclical gear configuration* than a solar train. See, e.g., ***Exhibit A***, Figs. 4 and 6. For at least these reasons, claim 7 does further limit claim 6, and does not fail for indefiniteness.

Upon review, Applicant submits that the Examiner will find all claims as presented in condition for allowance. Accordingly, a speedy Notice of Allowance is earnestly requested. Should any further issues remain, Applicant requests that the Examiner contact Applicant's representative at the telephone number listed below. To the extent any fees are due, their deduction is authorized from Deposit Account No. 11-0978.

Respectfully submitted,

KING & SCHICKLI, PLLC

Andrew D. Dorisio

247 North Broadway
Lexington, KY 40507
Telephone: (859) 252-0889

Exhibit A

United States Patent [19]

Murrell et al.

[11] **Patent Number: 4,774,855**

[45] **Date of Patent: Oct. 4, 1988**

[54] APPARATUS FOR PROVIDING AN ELECTRICAL GENERATOR WITH A CONSTANT ROTATIONAL SPEED FROM A VARIABLE SPEED INPUT

[75] Inventors: **Peter W. Murrell; John Calverley**, both of Cumbria, England; **Donald Williams; Douglas J. Thomas**, both of Powyf, Wales

[73] Assignee: **Vickers Shipbuilding and Engineering Limited**

[21] Appl. No.: **819,197**

[22] Filed: **Jan. 15, 1986**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 408,865, Aug. 17, 1982, abandoned.

[51] **Int. Cl.⁴** **F16H 47/04**

[52] **U.S. Cl.** **74/687**; 74/411; 74/720

[58] **Field of Search** .... 74/664, 681, 687, 705, 74/710, 720, 720.5, 411; 416/170 R; 290/44; 440/75

[56] **References Cited**

U.S. PATENT DOCUMENTS

3,298,251	1/1967	Moss	74/687
3,383,952	5/1968	Christenson	74/687 X
3,396,607	8/1968	Ross	74/687
3,534,631	10/1970	Keller	74/687
3,665,788	5/1972	Nyman	74/687 X
3,667,214	6/1972	Addie	74/411 X
3,714,846	2/1973	Louis et al.	74/687
3,776,796	10/1973	Henriot	74/411
3,943,787	3/1976	Hicks	74/411 X
3,964,334	6/1976	Hicks	74/411 X
3,983,764	10/1976	Hicks	74/411 X
4,040,508	8/1977	Sunada et al.	192/12 D
4,232,570	11/1980	Gibson	74/687 X
4,259,881	4/1981	Meyerle	74/687
4,341,132	7/1982	Burdick	74/687
4,353,269	10/1982	Hiersig	74/687 X
4,461,957	7/1984	Jallen	290/44
4,663,714	5/1987	Cornell et al.	74/687 X

FOREIGN PATENT DOCUMENTS

833273	4/1960	United Kingdom .
867316	5/1961	United Kingdom .
952982	3/1964	United Kingdom .
957573	5/1964	United Kingdom .
1097253	1/1968	United Kingdom .
1213178	11/1970	United Kingdom .
1381477	1/1975	United Kingdom .
1537730	1/1979	United Kingdom .
2042224	2/1980	United Kingdom .
1563698	3/1980	United Kingdom .
1601467	10/1981	United Kingdom .

OTHER PUBLICATIONS

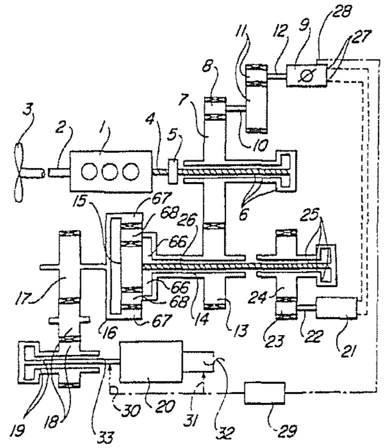
Allen Engineering Review, "Shaft-Alternators Driven Via Variable Ratio Epicyclic Gears", Poole; 11/1965.

Primary Examiner—Leslie A. Braun
Assistant Examiner—Dwight G. Diehl
Attorney, Agent, or Firm—Arnold S. Weintraub

[57] **ABSTRACT**

An electrical generator drive which comprises a power source able to transmit power at a rotational speed which may fluctuate substantially in service, an electrical generator which requires a rotational input of power at a substantially constant speed, and a controllable drive transmission of high rotational inertia coupling together the power source and the electrical generator. The drive transmission comprises an epicyclic gear having an input member driven by the power source, an output member coupled with the electrical generator, and a reaction member, a monitoring device responsive to fluctuations from a predetermined value in the rotational speed of the output member, and a control device including a hydraulic pump/motor unit coupled with the reaction member and controllable by the monitoring device in order to vary the relative rotation between the reaction member and the other members of the epicyclic gear so as to maintain a substantially constant predetermined speed of the output member. To enable the generator drive to respond rapidly to speed fluctuations of the power source, so as to maintain an substantially constant input speed to the generator, the monitoring device includes an electronic control unit having a memory which is programmed with a predetermined value corresponding to the required input speed to the generator derived from the output member. Further, in view of the high rotational inertia of the drive transmision, at least one flexible coupling is provided in the transmission to absorb torsionally any speed fluctuation imparted to the transmission by the power source.

8 Claims, 4 Drawing Sheets



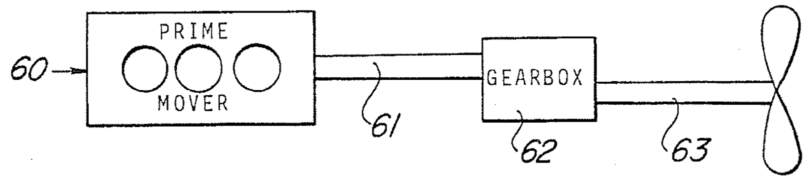


Fig-1

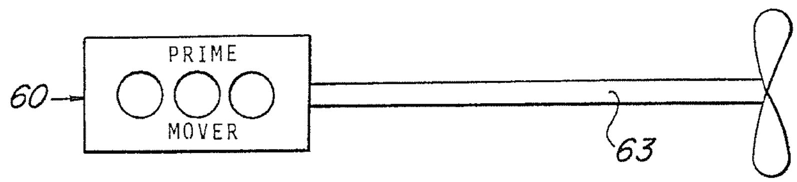


Fig-2

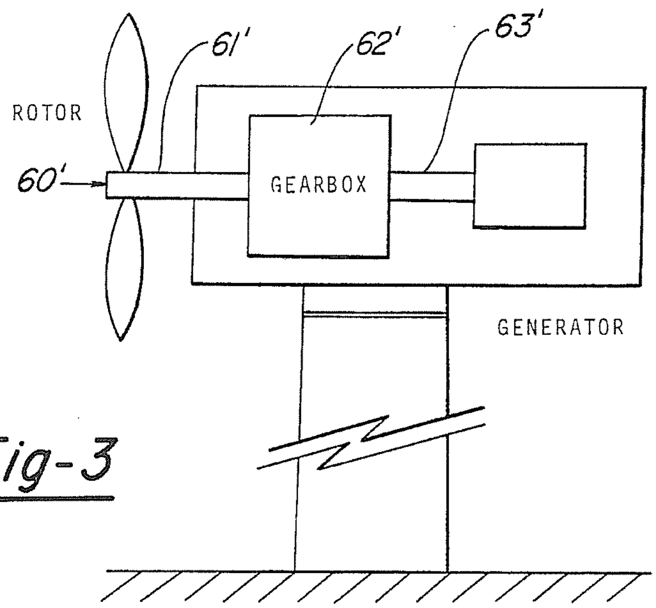


Fig-3

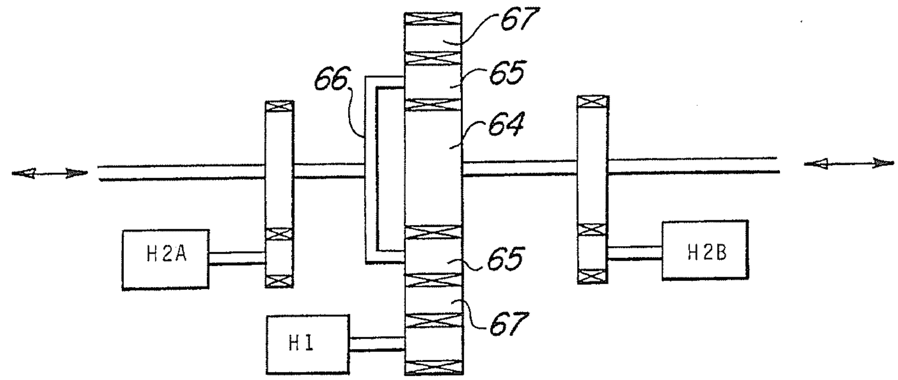


Fig-4

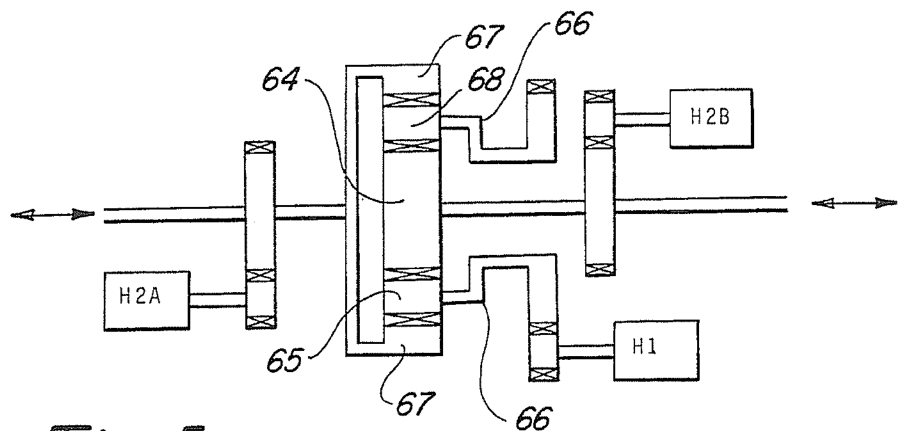


Fig-5

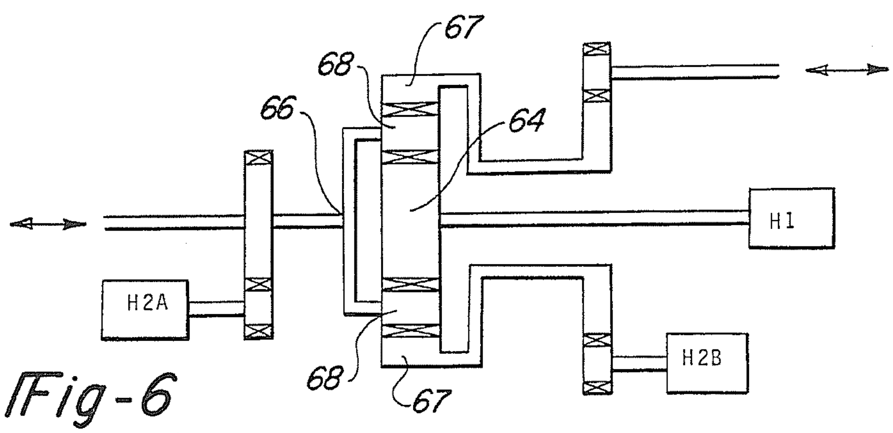


Fig-6

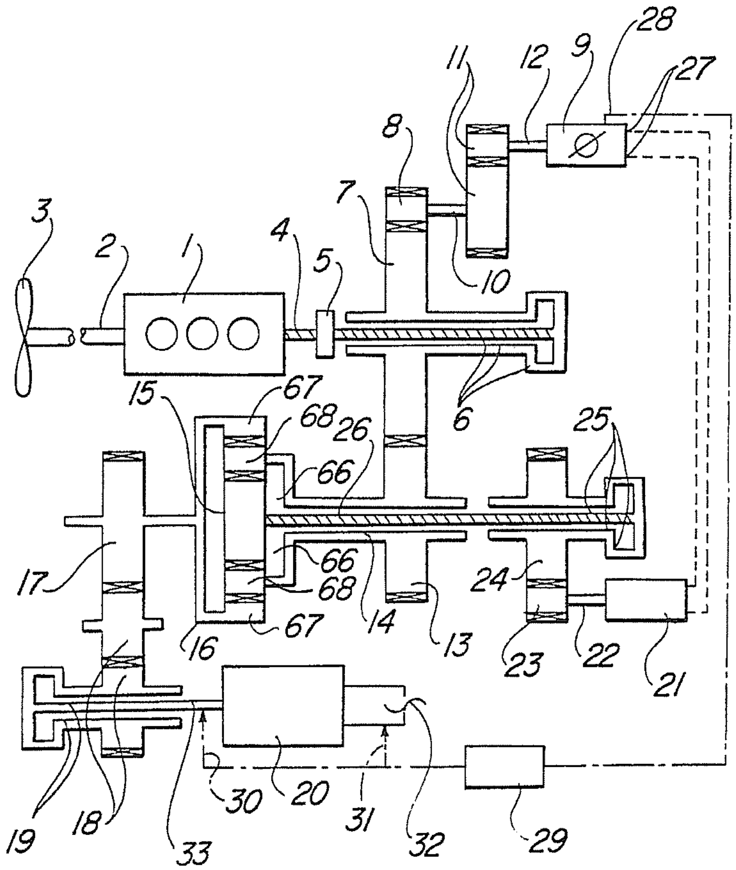

28
11
12
9
27
8
7
3
2
1
4
5
10
6
15
67
68
26
66
25
17
66
68
24
67
16
14
13
23
22
21
19
18
33
30
20
31
32
29

Fig-7

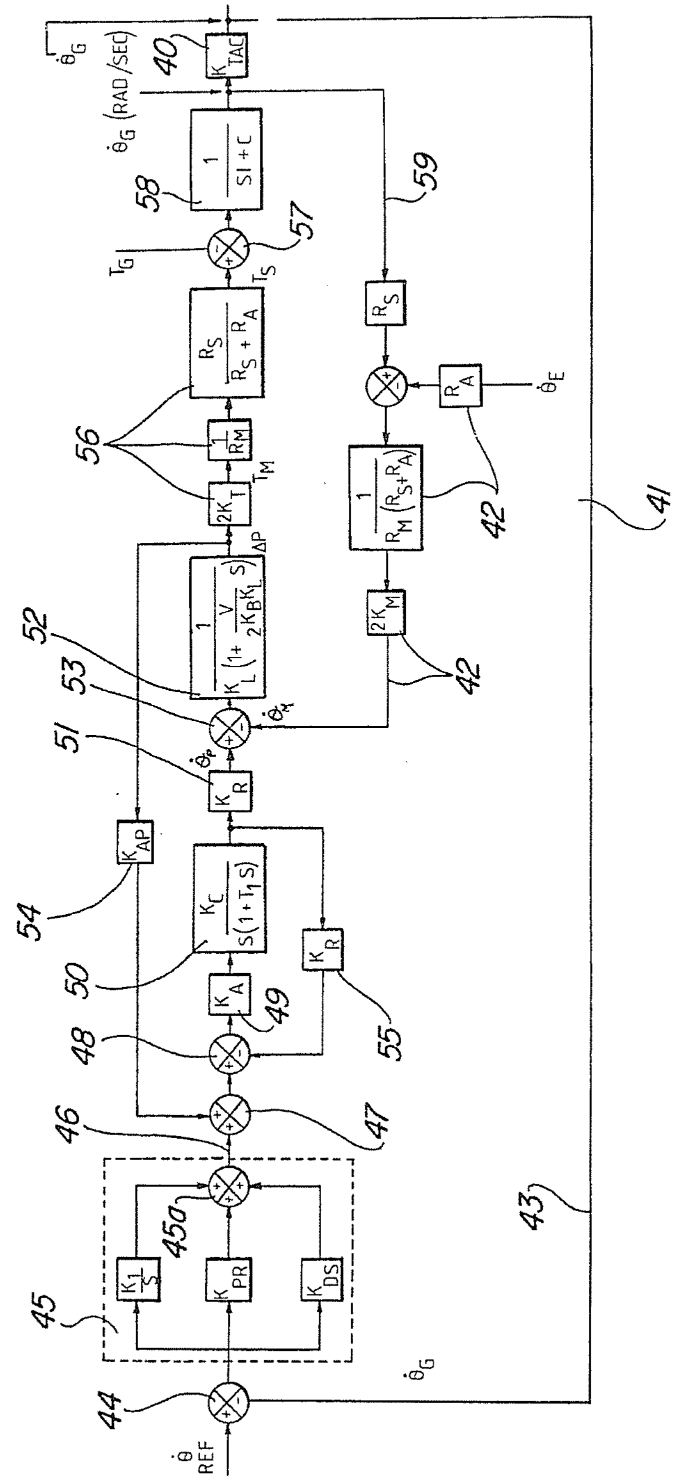


Fig-8

APPARATUS FOR PROVIDING AN ELECTRICAL GENERATOR WITH A CONSTANT ROTATIONAL SPEED FROM A VARIABLE SPEED INPUT

CROSS REFERENCE TO CO-PENDING APPLICATION

The subject application is a continuation-in-part application of co-pending application Ser. No. 408,865 (now abandoned) filed on Aug. 17, 1982 in the names of Peter W. Murrell, John Calverley and Donald Williams entitled APPARATUS FOR PRODUCING A CONSTANT ROTATIONAL SPEED FROM A VARIABLE SPEED INPUT.

BACKGROUND OF THE INVENTION

This invention relates to an electrical generator drive having a power source which is liable to transmit power at a rotational speed which may fluctuate substantially in service, an electrical generator which requires a rotational input of power at a substantially constant speed, and a controllable drive transmission of high rotational inertia coupling together the power source and the electrical generator.

1. Field of the Invention

The invention is particularly concerned with an electrical generator device which derives power from a power source which provides a speed output which can vary quite substantially in service, particularly the main engine on a marine vessel driving the propeller, though other examples of fluctuating power source with which the invention may be used include wind or water-powered devices. On a marine vessel, it is usual to employ an A.C. generator, but if the input speed to the generator varies, the frequency of the current produced will also vary, causing problems with some electrical equipment. For example, clocks will not keep correct time, and AC motors will run at variable speeds. Similarly, if a DC generator is used, the voltage of the current produced will also vary.

The generation of electrical power on board a ship can be derived from the prime mover of the ship, when it is not in port, since this is more economical than to operate a secondary auxiliary engine. However, a disadvantage of using the prime mover is that its speed, and that of the drive train, from the primer mover to an AC generator, may vary depending upon external factors e.g, if the ship is sailing through a storm. When a ship is cruising at a nominally constant speed, it might be expected that the engine(s) and propeller(s) would be rotating at a constant speed, but this is not always the case. Factors such as the roll of the ship, wave motion, changes in local water temperature and density, for example, can lead to a surprisingly large and random variation in the propeller speed. Thus, if a powered generator is driven via the prime mover drive train, any variation of the speed of rotation of the propeller shaft will be magnified by the gear ratio of a take-off drive to the generator, so that the rotational speed of the generator will vary through a greater magnitude speed range. In modern ships, alternating current power is generally used so that, if the speed of the AC generator varies, the current produced will have a variable frequency.

2. Description of the Prior Art

It has become common practice to electronically rectify the variable frequency to stabilize the frequency of the variable current produced from shaft driven generators, but for a large ship with a power requirements of between 0.5 and 1 MW, the physical size of the electronic equipment and the associated power loss pose severe problems for ship designers and operators.

Also, the generation of electrical power from natural sources, such as air or wind-driven devices, or water driven devices, e.g. turbines driven by tidal or other currents and devices arranged to extract energy from wave motion, usually suffers from the common disadvantage of fluctuation in the output speed of the device.

It is known from U.S. Pat. No. 3,298,251 to provide an electrical generator drive which is intended specifically for use with a turbo-jet engine. However, in this case the power source (the turbo-jet engine) will normally provide input power at a substantially constant speed, or at least the input speed to the generator drive will vary only gradually following operation of the throttle of the engine. Further, a hydraulic governor unit is provided in order to maintain a substantially constant input speed to the generator, and a governor unit of this type is particularly suitable for following progressive speed changes which would occur in service with a turbo-jet engine.

By contrast, the invention is concerned with providing an electrical generator drive which derives power from a power source which provides a rotational input of power to the generator which may fluctuate quite substantially in service, and any changes in speed are step changes, rather than progressive alterations in speed. Furthermore, the controllable drive transmission in an electrical generator drive according to the invention has a very high rotational inertia, and this magnifies the adverse effects of step changes in input speed to the generator drive from the power source.

SUMMARY OF THE INVENTION

According to the invention there is provided an electrical generator drive comprising a power source which is liable to transmit power at a rotational speed which may fluctuate substantially in service, an electrical generator which requires a rotational input of power at a substantially constant speed, and a controllable drive transmission of high rotational inertia coupling together the power source and the electrical generator, said drive transmission comprising:

an epicyclic gear having an input member which is arranged to be driven by said power source, an output member coupled with said electrical generator and a reaction member;

monitoring means arranged to respond to fluctuations from a predetermined value in the rotational speed of the output member;

and control means including a hydraulic pump/motor unit coupled with said reaction member and controllable by the monitoring means in order to vary the relative rotation between the reaction member and the other members of the epicyclic gear so as to maintain a substantially constant predetermined speed of the output member;

in which:

the monitoring means includes an electrical control unit having a memory storable with a predetermined value corresponding to the predetermined speed of the output member, means for feeding to the control unit an input signal which represents the actual speed of the output member and which the control unit compares with the predetermined speed of the output member, and a control line from the control unit to said pump-

/motor unit via which the control unit controls the operation of the pump/motor unit to cause alteration in the relative rotation of the reaction member, when a fluctuation occurs in the speed of the output member, and such as to restore the speed of the output member to a predetermined speed;

and the drive transmission also includes at least one flexible coupling for torsionally absorbing at least part of any speed fluctuation imparted to the drive transmission via the power source and the input member.

An electrical generator drive according to the invention is particularly suitable for use on board ship, in which the prime mover of the ship is used as the power source. However, other power sources may be employed, including air or wind driven devices, or water driven devices.

When the electrical generator drive is used on board ship, substantially constant frequency electrical power can be obtained by driving a marine generator from the output member of the epicyclic gear. Marine generators usually run at either 1200 or 1800 r.p.m., in order to generate 60 Hz electricity, in which case it is preferable to provide a variable ratio step-up drive when the ship is powered by a diesel engine. However, if the prime mover for the ship is a turbine, a variable ratio step-down drive could be more appropriate.

BRIEF DESCRIPTION OF THE DRAWING

FIG. **1** is a schematic illustration of a marine drive train which may be used as the power source for an electrical generator drive according to the invention, the drive train having a prime mover, a propeller shaft and a gearbox coupling together the prime mover and the propeller shaft;

FIG. **2** is a similar illustration of a marine drive train without a gearbox;

FIG. **3** illustrates a wind turbine drive train which may serve as an alternative power source for the electrical generator drive;

FIG. **4** is a schematic illustration of a variable ratio epicyclic gear, in a planetary arrangement, forming part of the drive transmission;

FIG. **5** is an illustration, similar to FIG. **4**, of an epicyclic gear in a star arrangement;

FIG. **6** illustrates an epicyclic gear in a solar arrangement;

FIG. **7** is a schematic illustration of an electrical generator drive according to the invention, incorporated in drive train of a ship; and

FIG. **8** is a schematic illustration of an electrical control unit employed in the electrical generator drive in order to cause automatic compensation for speed fluctuations of the power source in order to maintain a substantially constant input speed to an electrical generator.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings, FIGS. **1** and **2** show diagrammatically the basic layouts of conventional ship drive trains. FIG. **1** shows a prime mover, e.g. a slow or medium speed diesel engine or turbine, driving the propeller via a reduction gearbox. FIG. **2** shows a similar train in which the prime mover, e.g. a slow speed diesel engine, drives the propeller directly. In the drive train, shown in FIG. **1**, there are four possible power take-off points from which the apparatus may receive a fluctuating rotational input which are:

60—From the non-used end of the prime mover drive shaft.

61—From the shaft between the prime mover and gear box.

62—From the gear box.

63—From the propeller shaft.

In FIG. **2**, where there is no gear box, points A and D only apply.

FIG. **3** shows diagrammatically the drive train for an aerogenerator. The drive train comprises a wind turbine of viable speed coupled through the apparatus to a generator. The points of power source for the apparatus, equivalent to positions shown in FIG. **1**, are show as **61'**, **62'** and **63'**, and are:

61 In the shaft between rotor and gear box

62 In the gear box

63 In the gear box output shaft.

In this case, a second power take off from the rotor (**60**) (equivalent to position **60** in FIG. **1**) is not feasible.

An epicyclic gear consists of three basic elements which are the sun wheel, the carrier supporting a plurality of planetary pinions and an annulus gear; anyone of these basic elements may act as the input member, a second may act as the output member and the third will become the reaction member of the epicyclic gear. In order that the ratio of the epicyclic gear, in whatever configuration it is arranged, may be varied, the reaction member may be caused to revolve in either sense of rotation or be held stationary i.e. the relative rotation between the reaction member and the other members of the epicyclic gear may be varied. Thus for a constant, say positive, input speed of rotation, the ratio of the epicyclic gear mauy be infinitely and steplessly varied within given limits by turning the reaction member from its maximum negative speed of rotation, or vice versa through zero to its maximum positive speed of rotation.

FIGS. **4**, **5** and **6** show the basic planetary, star and solar epicyclic gear arrangement to achieve the variable-ratio gearing described in the preceding paragraph. In all of FIGS. **4**, **5** and **6**, solid arrows indicate the movement of power from left to right through the gear trains and this section of the specification is written on this basis; however, the power could equally well pass from right to left, as indicated by the dotted arrowheads. The elements of the epicyclic gear are marked for clarity as follows:

64 for sun wheel

65 for planet pinion

66 for the planet pinion carrier

67 for the annulus wheel

The reaction member of the epicyclic gear is the annulus wheel in the planetary arrangement (FIG. **4**), the planet carrier in the star arrangement (FIG. **5**), and the sun wheel in the solar arrangement (FIG. **6**). In each case, the reaction member is rotatably connected to a first hydraulic pump/motor **H1**, either directly as in FIG. **6** or indirectly via gearing as in FIG. **4** or indirectly via gearing and rotatable structural members as in FIG. **5**. Hydraulic pump/motors are usually used in pairs and in FIGS. **4**, **5** and **6**, the second hydraulic pump/motor may either be rotatably connected via gearing to the input shaft, where it is designated H2A, or to the output shaft where it is designated H2B. Hydraulic pump/motors **H1** and either H2A or H2B would be interconnected by two pipes (not shown for clarity) thus forming a closed loop around which hydraulic fluid may be circulated by either one acting as a

pump and causing the other to be operated as a motor. In the arrangement described, one of the pair of hydraulic pump/motors H1 or H2A (or H2B) would be fitted with a reversible, variable swash plate so that the quantity and direction of oil being circulated through the closed pipe loop and through the other of said pair of hydraulic pump/motors may be varied. By this means, the speed and sense of rotation of the reaction member may be varied infinitely between a given speed of anticlockwise rotation, through zero (i.e. stationary) to a given speed of clockwise rotation or vice versa. With such an arrangement and a constant speed drive to the input member, an infinitely variable speed may be obtained at the output member e.g. as in UK specification No. 1,097,253. The exact reverse of this is equally possible i.e. obtaining a constant speed output from a variable speed input and it is in this aspect of an epicyclic gear upon which the invention is based. It is, of course, also possible to obtain a variable speed output from a differently variable speed input, but the control problems in such a case are more complex.

The apparatus can receive a rotational input from a ships drive train at any point **60**, **61**, **62**, **63** in FIG. **1**, or **60** and **63** in FIG. **2**, or **61**, **62**, or **63** in FIG. **3**, to drive at constant speed, a generator via a variable ratio epicyclic gear train. The power required to operate the coupled pair of hydraulic motors controlling the speed and sense of rotation of the reaction member of the epicyclic gear is derived from the ships drive train. One hydraulic pump/motor, H1 is coupled directly to the reaction member, as in FIG. **6**, or via gearing to the reaction member as in FIGS. **4** and **5**.

FIG. **7** shows one preferred embodiment by which the principle disclosed herein may be put into effect. A prime mover **1**, for example a diesel engine, drives a propeller **3** via shaft **2**. At the other end of the prime mover, power is taken via a shaft **4**, a clutch or flexible coupling **5** and a cardan shaft/flexible coupling **6** to a gear wheel **7**. A pinion **8** meshes with gear wheel **7** and drives hydraulic pump/motor **9** via shaft **10**, step-up gearing **11** and shaft **12**. A second gear wheel **13** also meshes with gear wheel **7**. Gear wheel **13** is fast with a member **14** which also acts as the carrier **66** supporting the plurality of planetary pinions **65** of epicyclic gear **15**; thus the planet carrier **66** forms the input member of epicyclic gear **15** which, in this embodiment, is operated in the Solar configuration.

The output member of epicyclic gear **15** is the annulus wheel **67** which is rotationally fast with a member **16**. Gear wheel **17** is also rotationally fast with member **16** so that the output from epicyclic gear **15** passes via member **16** into gear wheel **17**, thence via step-up gearing **18**, cardan shaft/flexible coupling **19** and shaft **33** to generator **20**. The ratio of epicyclic gear **15** may be varied by controlled rotation of the sun wheel **64**. The speed and direction of rotation of the sun wheel **64** may be controlled by hydraulic pump/motor **21** which drives, via shaft **22**, a pinion **23** meshing with gear wheel **24**. Flexible coupling/cardan shaft **25** is rotationally fast with gear wheel **24** to transmit the drive via shaft **26** into sun wheel **64**. Shaft **26** is co-axial with gear wheels **24** and **13** and member **14**. Hydraulic pump/motor **21** is connected via piping **27** to pump/motor **9** to from a pair, of which one would have a fixed displacement and the other would have the reversible variable displacement swash plate.

Flexible couplings/cardan shafts **6**, **19** and **25** are typical of the sort of arrangements used to rotatably connect fixed members. Other types of coupling are equally possible. Also, a clutch which may be located in member **5** could equally well be located in the position of members **6**, **19** or **25**.

A control unit **29** is fed with an input signal which could either be a direct measure of the speed of rotation **30** of generator drive shaft **33** (or some other suitable shaft) or some property dependent upon the speed of rotation of generator shaft **33**, e.g. the frequency **31** of the alternating current **32** generated. Control unit **29** would compare the value of input signal **30** or **31** with the value stored in its memory and, if it differed by more than a defined amount, would produce an output signal **28** to alter the setting of the swash plate on hydraulic pump/motor **9**. This action would cause the flow of hydraulic fluid in pipes **27** to change so that speed of rotation (and possibly the direction of rotation as well) of pump/motor **21** would also change resulting in a new speed and/or direction of rotation of sun wheel **64** of epicyclic gear **15**. The new conditions now prevailing in the epicyclic gear **15** should restore the original speed of rotation of annulus wheel **67** and hence of generator shaft **33**.

It will be noted that gear wheels **7** and **13**, the configuration of epicyclic gear **15** and gear wheels **17** and **18** all give step-up ratios so that the speed of rotation gradually increases from, say, 65–100 RPM for a direct drive diesel engine or 400–800 RPM for a medium speed diesel engine driving through a gearbox to the 1200 or 1800 RPM required for a marine generator generating 60 Hz current. If the prime mover was a turbine operating at tens of thousands of revolutions per minute, the gear wheels **7**, **13**, epicyclic gear **15** and gear wheels **17** and **18** would have step-down ratios.

In a typical case, engine **1** may have a fuel efficient cruising speed of, say, P RPM and the gear ratios through the train would be so chosen that the correct alternator speed would be achieved with the reaction member of the epicyclic gear **15**, i.e. sun wheel **64**, not rotating. This would mean that pump/motor **21** was not rotating and that the swash plate of pump/motor **9** was set to zero so that no hydraulic fluid flowed in pipes **27**.

Consider now that the speed of prime mover **1** feel below P RPM. The whole train would slow down and input signal **30** or **31** would depart from the pre-set value stored in the memory of control unit **29**. This would cause control unit **29** to generate an output signal **28** dependent in magnitude on the difference between input signal **30** or **31** and the pre-set value stored in the memory. Output signal **28** would cause the swash plate control on pump/motor **9** to be moved by an appropriate means, e.g. a Servo mechanism (not shown) to, say, a positive setting. This would cause pump/motor **9** to act as a pump and cause hydraulic fluid to flow along pipes **27** thus making pump/motor **21** act as a motor and turn sun wheel **64** in a positive direction via the drive train consisting of members **22**, **23**, **24**, **25** and **26**. Thus there would be rotational input to epicyclic gear **15** via both sun wheel **64** and planet carrier **66** so that the speed of annulus wheel **67** would rise, as would the speeds of members **16**, **17**, **18**, **19** and **33** thus restoring the speed of generator **20** to its former level and input signal **30** or **31** to the pre-set value stored in the memory of control unit **29**. Having thus restored the input signal **30** or **31** to its pre-set value, the setting of the swash plate unit would remain fixed in that position until a further change in the speed of prime mover **1** occurred.

The operation of control unit 29 has been described in simple terms above to explain the basic principle. It is now necessary to consider the detailed operation of the system in relation to its environment, and detailed construction and operation of the control unit 29 will be described later with reference to FIG. 8 of the drawings. Reference will continue to be made to the marine application, though the arguments apply equally to other generators, whether driven by wave or wind power, etc.

Consider a ship at sea. Under normal conditions, the vessel will operate at a fuel-efficient cruising speed with only minor and progressive changes of engine speed. However, when storms occur, the ship will roll and pitch such that the propeller 3 will approach the surface of the sea and possibly partially, or even wholly, come out of the water. As the propeller approaches the surface, air bubbles will be formed due to cavitation, the resistance to rotation will fall and the rotational speed of the propeller will progressively rise. If the propeller breaks the surface of the sea the resistance to rotation will suddenly decrease and the speed of the propeller will rise in a stepwise fashion. As the propeller 3 is connected directly to the prime mover 1, the whole power train will be subject to the same rates of speed change.

Thus a control unit 29 is required which can react to both progressive and stepwise speed changes in the prime mover. Thus, the propeller 3 will be wholly or partially out of the water for only a relatively short time before the ship starts to pitch in the reverse direction causing it to be reimmersed. Thus there is great danger that a rapidly-varying input speed may cause the control unit 29 to "hunt" and set up oscillations in the speed of the components of the generator drive train. Oscillations in drive trains are never acceptable and particularly not when the magnitude of the rotational inertias involved is large. For example, for an 84,000 Tonne product tanker:

Mass of propeller =	35.1 Te
Reciprocating mass of engine =	63.825 Te
Max. normal speed of engine =	83 rpm
Rotational inertia of engine at 83 rpm =	266.551×10^3 kgm^2
Alternator rating =	1,500 kW
Alternator speed =	1,800 rpm
Rotational inertia of alternator at 1,800 rpm =	0.327×10^3 kgm^2

It will thus be evident that a highly sophisticated control system, capable of rapid response and able to minimize torque fluctuations in the gearing trains, is needed. A complex electronic control system as shown in FIG. 8 must be used to achieve these requirements. Conventional hydraulic control systems are too slow in their operation and can lead to "hunting" when subject to rapid speed fluctuations. Simple electronic control systems may have a rapid enough response time, but "hunting" is also likely to be a problem. The system shown in FIG. 8 can provide the rate of response needed without "hunting" occurring. However, even with such a sophisticated system, torque fluctuations will occur in the gear train. This is why torsional flexibility is incorporated through the drive train from prime mover 1 to generator 20 via flexible coupling 5, and cardan shafts 6, 19 and 25.

FIG. 8 is a transfer function block diagram of the control system. This is an analogue model of the system and is used to show the parameters of the system and their interactions when changes occur. It must also be borne in mind that the control means is a dynamic system and, when required, all react to a continuously varying input.

In FIG. 8, the symbols used have the following means:

KI	INTEGRAL GAIN	
KPR	PROPORTIONAL GAIN	P.I.D.
KD	DERIVATIVE GAIN	CONTROLLER
KA	SERVO AMPLIFIER GAIN	
KC	SWASH GAIN	
TI	SWASH TIME-CONSTANT	
KR	SWASH POTENTIOMETER GAIN	
KP	HYDRAULIC PUMP FLOW CONSTANT	
KΔP	DIFFERENTIAL PRESSURE FEEDBACK GAIN	
KL	CO-EFFICIENT OF HYDRAULIC LEAKAGE	
V	VOLUME OF HYDRAULIC FLUID UNDER COMPRESSION	
KB	BULK MODULUS OF HYDRAULIC FLUID	
KT	HYDRAULIC MOTOR TORQUE CONSTANT	
KM	HYDRAULIC MOTOR FLOW CONSTANT	
KTAC	TACHOGENERATOR GAIN	
RA	ANNULUS RADIUS	CONTROL EPICYCLIC
RS	SUN RADIUS	
RM	GEAR RATION, HYDRAULIC MOTORS TO CONTROL CARRIER	

Assume that the system has been operating at a constant engine speed and is in equilibrium, with the generator rotating at the desired 1800 rpm. In this case, the signal from the tachometer 40 $\dot{\theta}_G$, is a constant value equivalent to 1800 rpm. Similarly the engine speed $\dot{\theta}_E$ is constant. (N.B. The engine speed does not have to be measured by the control unit 29, but it is included in the analogue logic).

Consider now that the engine speed $\dot{\theta}_E$ 41 falls. This will have a double adverse effect as the speed of pump-/motor 9 will fall via gearing 8 and 11, and the speed of the carrier of epicyclic gear 15 will fall via gearing 7 and 13. Thus as the speed of rotation of both input members of epicyclic gear 15 falls, the speed of the output annulus member of epicyclic gear 15 will also fall but by a greater amount due also to the effect of the epicyclic gear ratio. This double adverse effect is fed into the system via logic elements 42 as a signal $\dot{\theta}_M$. As the speed of the annulus member of epicyclic gear 15 falls, so does the speed of generator 20 causing the tachometer 40 to produce a (negative signal $\dot{\theta}_G$, which is passed via connection 43 to comparator 44. $\dot{\theta}_G$ is compared with the reference value $\dot{\theta}_{ref}$ stored in the comparator's memory and a signal representing the sign and magnitude of the difference between $\dot{\theta}_G$ and $\dot{\theta}_{ref}$ is passed to P.I.D. controller 45. P.I.D. controller 45 contains integral, proportional and derivative gain components to determine what is actually occurring to the tachometer reading and its rate of change. Inside the P.I.D. controller 45, a comparator 45A assess the three components and produces the correct output signal 46. The integral, proportional and differential gains of P.I.D. controller 45 are preset for the required accuracy of control and speed of response.

Signal 46 is passed via two further comparators 47 and 48 and thence to amplifier 49 which sends an amplified signal (not shown) to alter the swash plate angle. Blocks 50, 51 and 52 represent the swash plate dynamic (gain and time constant including servo control), pump flow constant and properties of hydraulic fluid, etc.,

respectively. Two feed back loops are employed in this part of the analogue model. The differential pressure feedback **54** allows for the change in leakage in the hydraulic pump/motors due to the changes in swash plate angle and load; this is an important factor as it stops "hunting". The potentiometer feedback **55** is a critical factor in achieving the required speed of response as it ensures the swash plate is moved to the new setting as quickly as possible and then rapidly damps any resulting fluctuations. Both feedback signals **54** and **55** are fed into comparators **47** and **48** respectively to modify signal **46** before it is amplified **49** and sent to the swash plate servo (not shown).

The change in the flow of hydraulic fluid from pump/motor **9**, $\dot{\theta}_M$, is introduced into the analogue logic via element **53**.

From element **52**, the analogue model then incorporates the motor torque constant, intermediate gearing and epicyclic gearing ratios **56**. The electrical load torque, T_G on generator **20**, is represented as an input **57** and by block **58**. As the load demanded from generator **20** varies, the power input in shaft **33** will vary in a similar manner; variation in power transmission in the constant speed drive will affect other parts of the power train, particularly if part of the power of the epicyclic gear is passing via the hydraulic path. The torque T could be measured if required and used as an input to the control system **29**. However, in practise this is not necessary, as the changing load would cause speed variations in the generator, which would be recorded by the tachometer **40**; this effect is shown by connection **59**.

The control system described above uses only one input measurement, i.e. the speed of rotation of generator **20** or the frequency of the output current (60 Hz), yet it can accommodate external influences, such as variations in engine speed or electrical demand, and internal influences, such as variations in hydraulic fluid flows, swash plate angle, leakage as well as the various gains, gear ratios, time constants and physical properties of the components of the system. Despite these influences, the control system has been proved to maintain the frequency of the power generated at 60 Hz±1%, with the ship's engine varying in speed between 58 and 83 rpm; this satisfies the requirements of Lloyds Rules and Regulations.

In practice on a ship there would be at least two generators. One would be driven from the prime mover **1** for use at sea. A second would be driven by a separate diesel engine for use in port. "Droop Control" (not shown) is incorporated into the control system, described above, to permit load sharing and load transference between the two (or more) generators.

Returning now to the mechanical functions of the generator drive train, there are two power paths to the epicyclic gear **15** which are:

(i) direct power into the planet carrier C via the mechanical path formed by members **4, 5, 6, 7, 13** and **14.**

(ii) indirect power into the sun wheel **64** via the two hydraulic pump/motors **9, 21** and interconnecting piping **27** in a non-mechanical path.

Both the above paths put power into the epicyclic gear **15** the output from which, via annulusgear **67**, drives the generator.

If the speed of prime mover **1** now rises to a value greater than P RPM, the whole train will speed up accordingly and input signal **30** or **31** would again depart from its pre-set value as stored in the memory of the control unit **29**. In this instance the sign of the difference between the magnitudes of input value **30** or **31** and the value in the memory would be different to that previously so that the output signal **28** generated by control unit **29** would cause the setting of the swash plate of pump/motor **9** to be moved to a negative position. Thus, hydraulic fluid will flow in the opposite direction along pipes **27** and pump/motor **21** and sun wheel **64** of epicyclic gear **15** will both rotate in the negative sense. In this case, pump/motor **21** would act as a pump and pump/motor **9** would act as the motor. Here the two power paths to the epicyclic gear become:

(i) direct power into the planet carrier **66** via the mechanical path formed by members **4, 5, 6, 7, 13** and **14.**

(ii) indirect power out of the sun wheel **64** via the two hydraulic pump/motors **9, 21** and interconnecting piping **27** in a non-mechanical path. In this case, power is passed into the epicyclic gear **15** via the planet carrier **66** and out via the annulus gear **67** to drive the generator, with the excess power being removed via the sun wheel **64**. Thus a partial power recirculation loop is set up via members **15S, 26, 25, 24, 23, 22, 21, 27, 9, 12, 11, 10, 8** back to gear wheel **7**.

It is also possible to use a hydraulic pump/motor **9** with only a variable swash plate i.e. not a reversible one. In this case, the swash plate of pump/motor **9** could be set to an intermediate point of its travel when prime mover **1** was operating at its cruising speed of P RPM. If the speed of the prime mover fell below P RPM, the angle of the swash plate of pump/motor **9** would be increased so that the speed of pump/motor **21**, and hence also of sun wheel of epicyclic gear **15**, would increase to compensate for the reduction in speed of member **14**. Conversely, if the speed of the prime mover rose above **65**, the angle of the swash plate of pump/motor **9** would be reduced to reduce the speed of pump/motor **21** and sun wheel of epicyclic gear **15**. When a non-reversible swash plate pump/motor **9** is used, power is always put into epicyclic gear **15** via both the mechanical and hydraulic paths and no power recirculation loop is possible.

As the power carrying capacity of hydraulic pump/motors is limited, installations for the generation of substantial quantities of electrical power would be likely to prefer a reversible variable swash plate hydraulic pump/motor **9**, so that only a minimum amount of power would be transmitted in the non-mechanical path.

Though the embodiment shown in FIG. **7** may at first sight seem complicated, the arrangement is designed for high reliability and easy access for maintenance in the confined space of a ship's engine room. Further reductions of space and increase in reliability may be obtained if the power in epicyclic gear **15** is carried uniformly by all the planetary pinions **65**. This can be achieved by the use of load sharing elements to support the planetary pinions **65** from the planet carrier **66** such as flexible pins as disclosed in more detail in U.K. Patent Specification No. 1,101,131.

The purpose of this disclosure is to so control the speed of the reaction member of epicyclic gear **15** that a constant speed output may be obtained. This control has been described using hydraulic means, but could equally well be performed by a variable speed or reversible and variable speed electric or pneumatic motor.

Although not illustrated, natural power sources may be used to drive the apparatus which have fluctuating

outputs, such as air or wind-driven devices, water driven devices, and devices which take energy from wave-motion as described for example in U.K. Patent Specification No. 1,601,467.

We claim:

1. An electrical generator drive having a power source which is able to transmit power at a rotational speed subject to progressive and stepwise speed changes, an electrical generator which requires a rotational input of power at a substantially constant speed, and a controllable drive transmission of high rotational inertia coupling together the power source and the electrical generator, said drive transmission comprising:

an epicyclic gear having an input member which is arranged to be driven by said power source, an output member coupled with said electrical generator, and a reaction member;

monitoring means arranged to respond to fluctuations from a predetermined value in the rotational speed of the output member;

and control means including a hydraulic pump/motor unit coupled with said reaction member and controllable by the monitoring means in order to vary the relative rotation between the reaction member and the other members of the epicyclic gear so as to maintain a substantially constant predetermined speed of the output member;

in which:

the monitoring means includes an electrical control unit having a memory storable with a predetermined value corresponding to the predetermined speed of the output member, means for feeding to the control unit an input signal which represents the actual speed of the output member and which the control unit compares with the predetermined speed of the output member, and an electrical control line from the control unit to said pump/motor unit via which the control unit controls the operation of the pump motor unit to cause alteration in the relative rotation of the reaction member, when a fluctuation occurs in the speed of the output member, and such as to restore the speed of the output member to the predetermined speed;

and the drive transmission also includes at least one flexible coupling for torsionally absorbing at least part of any speed fluctuations imparted to the drive transmission via the power source and the input member.

2. An electrical generator drive according to claim 1, in which the power source is the prime mover of a marine vessel.

3. An electrical generator drive according to claim 2, in which the drive transmission comprises an input shaft which is coupled with the prime mover, a cardan shaft coupled with the input shaft via a first flexible coupling, a gear wheel rotatable with the carden shaft and coupled with the input member of the epicyclic gear, a further gear wheel rotatable with the output member of the epicyclic gear, and a further cardan shaft arranged to drive the electrical generator and coupled with said further gear wheel.

4. An electrical generator drive according to claim 1, in which the electrical generator is an A.C. generator, and the monitoring device is arranged to monitor the frequency of the A.C. generator.

5. An electrical generator drive according to claim 1, in which the control means comprises a linked pair of first and second hydraulic pump/motors, one of which is of variable displacement, and said first pump/motor being arranged to be driven directly by the drive transmission and said second pump/motor being coupled with said reaction member of the epicyclic gear, and in which said monitoring means is operable to control the setting of the variable displacement pump/motor so that said second pump/motor can vary the relative rotation between the reaction member and the other members of the epicyclic gear and thereby maintain the substantially constant predetermined output speed of the output member, when fluctuation occurs in the speed of the input member.

6. An electrical generator drive according to claim 1, in which the power source comprises a fluid-driven device.

7. An electrical generator drive according to claim 6, in which the fluid-driven device comprises a rotary wind-driven device.

8. An electrical generator drive according to claim 1 wherein said members of said epicyclic gear are constituted by a sun wheel, a carrier supporting a plurality of planetary pinions each of which is in driving engagement with the sun wheel, and an annular gear in driving engagement with said planetary pinions.

* * * * *

Exhibit B

United States Patent [19]

Hansson

[11] **4,104,932**

[45] **Aug. 8, 1978**

[54] **EPICYCLIC GEAR TRAINS**

[75] Inventor: **Hans-Erik Hansson**, Finspong, Sweden

[73] Assignee: **Stal-Laval Turbin AB**, Finspong, Sweden

[21] Appl. No.: **740,522**

[22] Filed: **Nov. 10, 1976**

[30] **Foreign Application Priority Data**

Nov. 10, 1975 [SE] Sweden 7512553

[51] **Int. Cl.**[2] **F16H 1/28**; F16H 57/00

[52] **U.S. Cl.** **74/801**; 74/410

[58] **Field of Search** .... 74/410, 411, 801, 750 R

[56] **References Cited**

U.S. PATENT DOCUMENTS

1,388,897	8/1921	Schmidt	74/410
2,127,463	8/1938	Chilton	74/801
2,233,498	3/1941	Taylor	74/801
2,543,453	2/1951	Fuller	74/750 R X
2,749,778	6/1956	Kuhn	74/801
3,303,713	2/1967	Hicks	74/801
3,776,067	12/1973	DeBruyne et al.	74/801
3,943,787	3/1976	Hicks	74/410
3,964,334	6/1976	Hicks	74/801 X
3,983,764	10/1976	Hicks	74/410

FOREIGN PATENT DOCUMENTS

2,361,614 6/1974 Fed. Rep. of Germany.

Primary Examiner—Samuel Scott
Assistant Examiner—Lance W. Chandler
Attorney, Agent, or Firm—Pollock, Vande Sande & Priddy

[57] **ABSTRACT**

An epicyclic gear train comprises support spindles for the planet or star wheels cantilevered from a single support flange. The cross sections of the spindles vary with distance from the support plate so that under tangential loading the bearing surfaces of the spindles will assume a position essentially parallel to the axis of the gear train.

7 Claims, 2 Drawing Figures

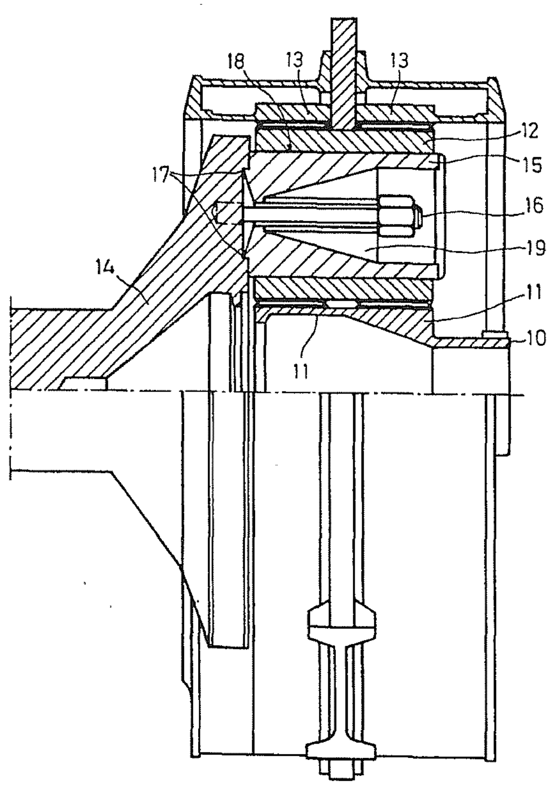


Fig.1

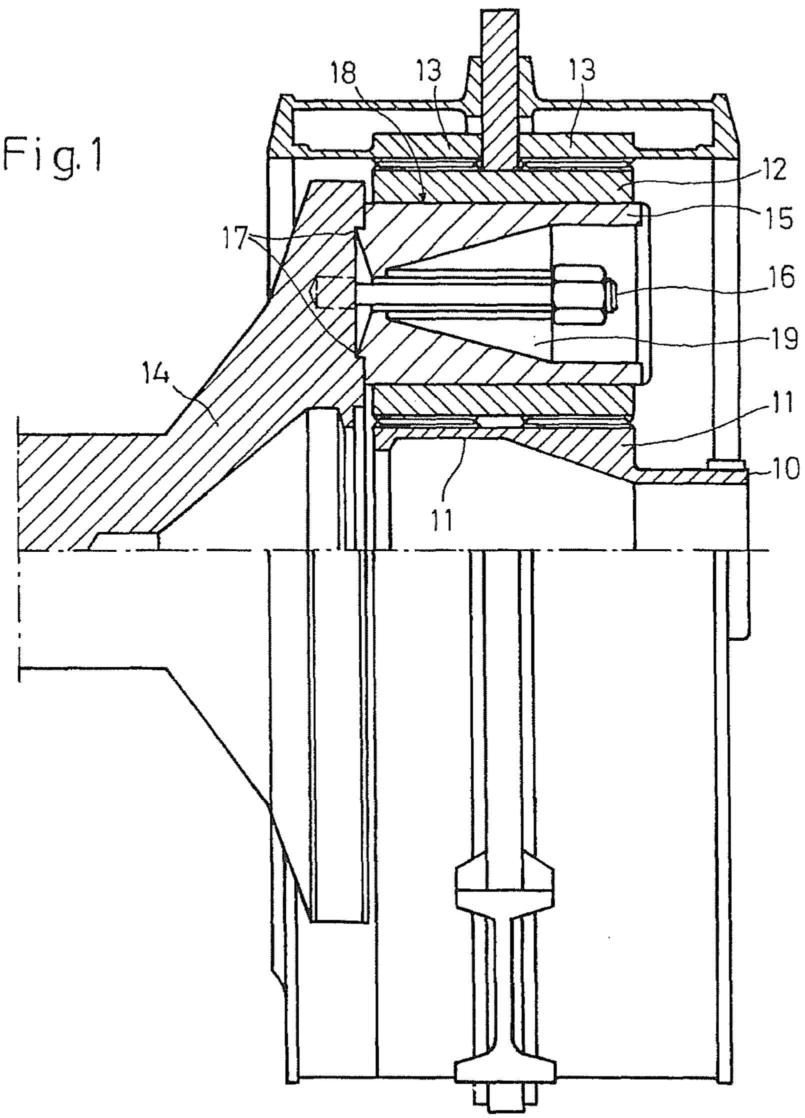
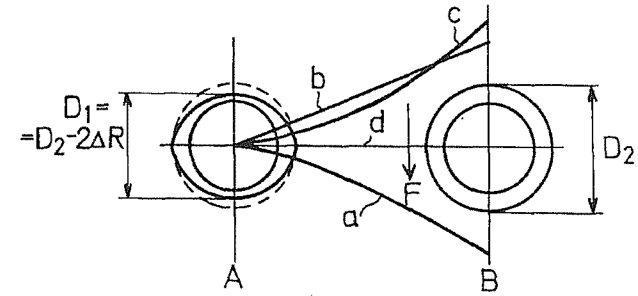


Fig.2



EPICYCLIC GEAR TRAINS

BACKGROUND OF THE INVENTION

In the prior art, the holder for the star or planet wheels of an epicyclic gear train typically comprises two parallel flanges between which the star or planet wheels are journalled on shaft pins. In a planetary gear train, one of these flanges is torque-transmitting since it is connected for rotation with an output shaft; whereas, in a star gear train, the holder is fixedly anchored in the gear housing and torque is transmitted by the inner toothed ring gear of the gear train. In either case, the two flanges are rigidly connected by means of bars; but no matter how strong this connection is made a certain deformation of the planet wheel holder cannot be avoided. Such deformation results in a certain binding or clamping between on the one hand, the planet wheels and, on the other, the shaft pins, sun wheels and surrounding gear ring. In this connection, it should be noted that the various parts of the planet wheel holder cannot be increased in size without limit in order to minimize deformation since the dimensions of the planet wheel holder are constrained by the space available between the different wheels of the gear train.

OBJECTS OF THE INVENTION

An object of the invention is to provide a single flange support for the planet wheels of a planetary gear train.

A further object of the invention is to provide a mounting spindle for such planet wheels which is specially configured to ensure that the planet wheels turn on an axis essentially parallel to the axis of the gear train.

Still another object of the invention is to provide such a mounting spindle having attachment bosses for facilitating installation in proper orientation with such a single flange holder.

These objects are given only by way of example. Thus, other desirable objects and advantages inherently achieved by the invention may occur to those skilled in the art. Nonetheless, the scope of the invention is to be limited only by the appended claims.

SUMMARY OF THE INVENTION

With an epicyclic gear according to the present invention, the holder can be made with a single flange, so that double flanges and bars are completely avoided. In the invention, the planet or star wheels due to their unique construction and mode of attachment to the single holder, compensate for elastic deformations arising in the wheel spindles.

BRIEF DESCRIPTION OF THE DRAWING

The epicyclic gear train according to the invention will be described in more detail with reference to the accompanying drawing, in which:

FIG. **1** shows a partially sectional view of an epicyclic gear train according to the invention with a single flange holder; and

FIG. **2** illustrates by graphical constructions the geometrical embodiment of a wheel spindle according to the invention indicating cross sections of the planet wheel spindles at their opposite ends.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. **1** shows an epicyclic gear, more specifically a planetary gear having sun wheel shaft **10**, sun wheel **11**, planet wheels **12** (only one shown), inner toothed rings **13** and a planet wheel holder **14**. Each planet wheel **12** rotates on a spindle **15** attached to the holder **14** by means of a set bolt **16** arranged centrally in the spindle. Because of its central positioning the bolt **16** transmits no tangential force, but only axial, attachment force. The tangential force acting on the spindle **15** is taken up by guides **17** arranged in the holder **14**. Guides **17** are recesses shaped to receive a correspondingly shaped boss extending from spindle **15** to lock spindle **15** in a fixed orientation relative to holder **14**.

Spindle **15** varies in cross section from its end near holder **14** to its outer end so that the shape of bearing surface **18** changes with distance from holder **14**. The section is not entirely circular right next to holder **14** as shown at section A in FIG. **2**, and gradually tapers outward to become circular as shown at section B in FIG. **2**. The cross section axes shown at A and B are preferably tangential to the bolt circle of bolts **16**; however, due to rotational forces and possible assymmetry in the gear train, these axes may deviate somewhat from their preferred orientation. The gradually changing cross section of spindle **15** provides compensation for the deformation which would occur if the cross section of spindle **15** were a right circular cylinder extending from unilateral holder **14**. Deformation of a right circular cylindrical spindle would result in an unacceptable unbalance of the associated planet wheel and, as a result of the deformation, poor utilization of the bearing surface between the spindle and the planet wheel. For example, if spindle **15** were a right circular cylinder along its entire bearing surface, as indicated by the broken lines in section A, it would be bent by a tangential force F acting parallel to axes A and B on the spindle, so that bearing surface **18** between spindle **15** and the planet wheel **12** would deform approximately as shown by curve *a*, where the line *d* indicates the center line of the unloaded spindle.

If, on the other hand, the shape of bearing surface **18** of spindle **15** is varied in accordance with the invention, preferably by a linear transition from section A to section B as shown by curve *b*, the deformation caused by tangential forces is compensated. When surface **18** has an unloaded contact surface having the shape of curve *b*, the surface will deform under loading to a curve which closely parallels line *d* and hence the axis of the gear train itself. Alternatively, a curve according to *c* can be formed on bearing surface **18** by applying a tangential force F to prestress the work piece during machining of the bearing surface, for example, so that the bearing is better utilized. Curve *c* would thus be inverse of curve *a*.

The cross section of spindle **15** at section A preferably comprises two circular segments each having a radius of curvature equal to that of the circular cross section at section B. These segments intersect at two diametrically opposed points each spaced from line *d* a distance slightly less than the radius of the circular cross section at section B, the points also lying in a plane passing through the line *d* perpendicular to axes A and B. The bearing surface then tapers to the circular shape shown at section B. This taper changes the cross section of the spindle with distance from the holder so that

under tangential loading, the side of the spindle supporting the planet wheel will deform and yet have a bearing surface which is essentially cylindrical parallel to the unloaded line *d* where the spindle bears on the surface. So, depending on the direction of rotation of holder **14**, planet wheel **12** rides on one or the other of the two circular segments at section A. Thus, the planet wheels run parallel to line *d*, which minimizes binding or clamping among the various planet wheels, shaft pins, sun wheels and inner toothed gear ring. Of course, other cross sections could be used at section A without departing from the scope of the invention, so long as the bearing surface between the spindle and the planet wheel deforms to be essentially circular and approximately parallel to the line *d*, as taught by this invention.

In FIG. **2** the difference $2\Delta R$ between the dimensions D_1 and D_2 of the spindle in the two sections A and B has been greatly exaggerated in order to illustrate the principle of the invention. A realistic value for ΔR is about 1/10 mm when D_2 is 200–500 mm.

As shown in FIG. **1**, the spindle is further provided with an internal recess **19** having a decreasing diameter towards the part of the spindle which is secured to the holder **14**, since the bending stress and thus the deformation will be greater in this part of the spindle. The recess **19** has the effect of decreasing the mass of the spindle with distance from the holder **14**.

I claim:

1. An epicyclic gear train, comprising:

a plurality of planet or star wheels;

a single flange holder for said wheels;

a plurality of spindles extending from said holder and supporting said wheels for rotation on said spindles, each of said spindles having an unloaded axis, a first cross section at the end of said spindle adjacent said holder and a second different, cross section at the end of said spindle remote from said holder, each of said spindles also having a bearing surface for supporting said wheels, said bearing surface extending between said first and second cross sections, and the cross sectional area of said spindle gradually changes therebetween whereby as said spindle deforms under tangential loading, said bearing surface between said spindle and its associated wheel deforms until it is essentially parallel to said unloaded axis.

2. Apparatus according to claim **1**, wherein said bearing surface tapers linearly from said first cross section to said second cross section.

3. Apparatus according to claim **1**, wherein said bearing surface tapers non-linearly from said first cross section to said second cross section.

4. Apparatus according to claim **1**, wherein said second cross section is circular and said first cross section has at least two segments of equal radius with said second cross section.

5. Apparatus according to claim **1**, wherein each of said spindles is affixed to said holder by means of a set bolt arranged centrally in each of said spindles.

6. Apparatus according to claim **1**, wherein said holder comprises guides for receiving said spindles and taking rotary-tangential forces acting on said spindles.

7. Apparatus according to claim **1**, wherein said spindle comprises means for reducing its mass as a function of distance from said holder.

* * * * *

Exhibit C

US006223616B1

(12) **United States Patent**
Sheridan

(10) **Patent No.: US 6,223,616 B1**
(45) **Date of Patent: May 1, 2001**

(54) **STAR GEAR SYSTEM WITH LUBRICATION CIRCUIT AND LUBRICATION METHOD THEREFOR**

(75) Inventor: **William G. Sheridan**, Southington, CT (US)

(73) Assignee: **United Technologies Corporation**, Hartford, CT (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/470,230**

(22) Filed: **Dec. 22, 1999**

(51) **Int. Cl.**[7] **F16H 57/04**
(52) **U.S. Cl.** **74/468**; 475/159; 184/6.12
(58) **Field of Search** 475/159; 74/467, 74/468; 184/6.12

(56) **References Cited**

U.S. PATENT DOCUMENTS

2,591,743		4/1952	Thompson	475/346
2,684,591		7/1954	Lundquist	73/862.31
2,703,021		3/1955	Stoeckicht	74/410
2,749,778		6/1956	Kuhn	475/347
3,160,026		12/1964	Rosen	74/410
3,307,433		3/1967	Bennett et al.	475/342
3,352,178		11/1967	Lindgren et al.	475/343
3,650,353		3/1972	Abbott	184/6.12
3,776,067		12/1973	DeBruyne et al.	475/159
3,960,029	*	6/1976	Eichinger	184/3.12 X
4,271,928		6/1981	Northern	184/6.4
4,378,711		4/1983	Daniel	74/467
4,583,413	*	4/1986	Lack	74/467 X
5,102,379	*	4/1992	Pagluica et al.	475/159 X
5,433,674	*	7/1995	Sheridan et al.	475/346
5,466,198	*	11/1995	McKibbin et al.	475/346
5,472,059	*	12/1995	Schlosser et al.	475/159 X
5,472,383		12/1995	McKibbin	475/159
5,643,126	*	7/1997	Hotta et al.	475/159
5,685,797	*	11/1997	Barnsby et al.	475/331
5,830,098	*	11/1998	Kimes	475/159

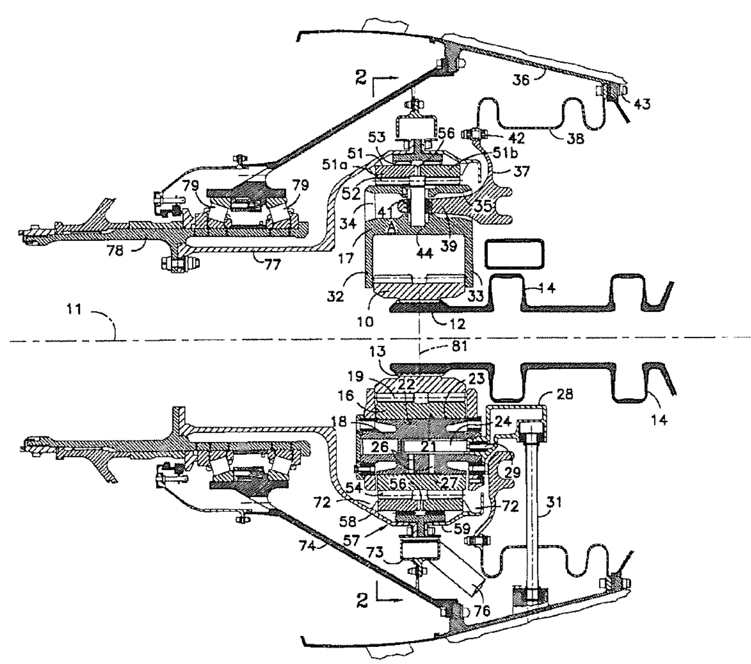
* cited by examiner

Primary Examiner—Charles A Marmor
Assistant Examiner—Ankur Parekh
(74) *Attorney, Agent, or Firm*—Kenneth C. Baran

(57) **ABSTRACT**

A star configured epicyclic gear train includes a sun gear **10**, a plurality of star gears **16** supported on bearings **18**, a ring gear **51** and a set of baffles **44** disposed between the star gears. A single lubricant circuit serves as the exclusive means for supplying lubricant successively to the bearings, the sun/star mesh and the star/ring mesh. A variant of the gear system includes an auxiliary lubricant circuit featuring a set of spray bars **82** for supplying lubricant to the sun/star mesh and the star/ring mesh. The primary and auxiliary circuits share a common lubricant discharge path representing the exclusive means for evacuating both the primary and auxiliary lubricant from the gear system. Preferably, the auxiliary lubricant circuit is selectively operable. The inventive gear system addresses a number of difficulties related to lubricating and evacuating lubricant from a star configured epicyclic gear system.

15 Claims, 5 Drawing Sheets



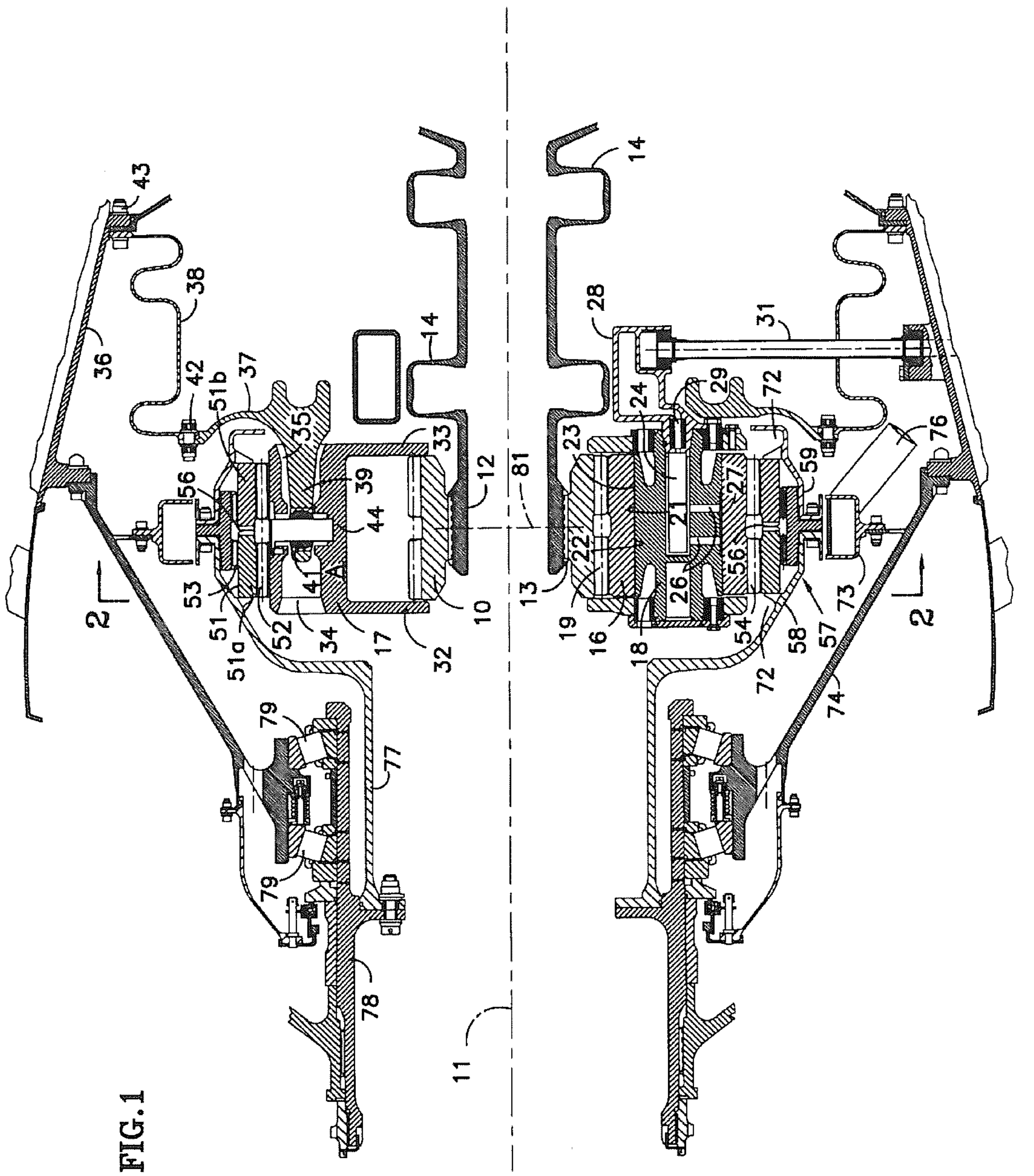
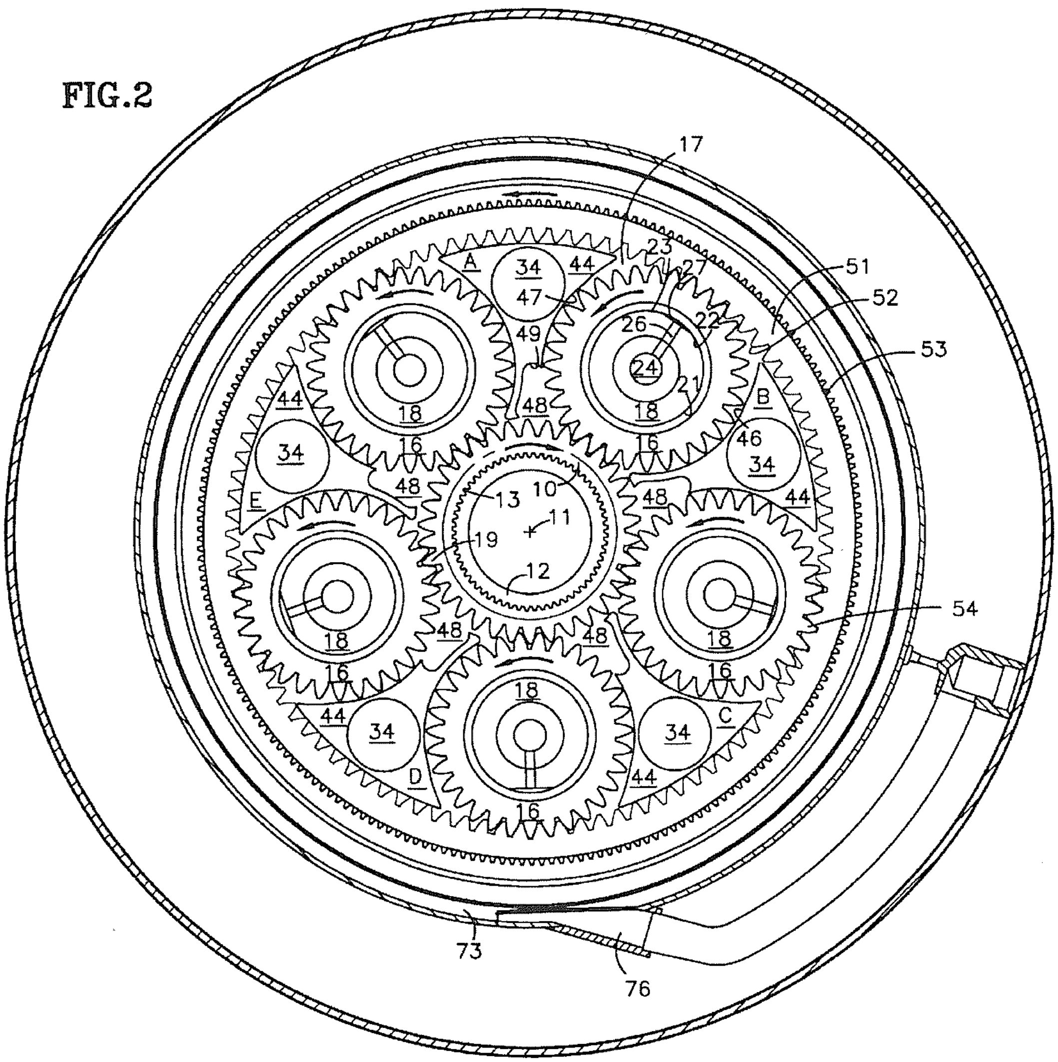

43
38
36
42
51b
37
35
14
33
12
81
39
44
56
53
51
51a
52
34
41
17
32
10
2
79
77
79
78
11
14
28
31
24
29
72
23
59
76
27
21
22
13
19
16
18
26
56
54
72
58
57
73
2
74
FIG. 1

FIG.2


17
23
27
26
22
24
21
51
52
53
A
B
C
D
E
34
44
47
49
48
18
16
46
13
10
11
19
12
54
73
76

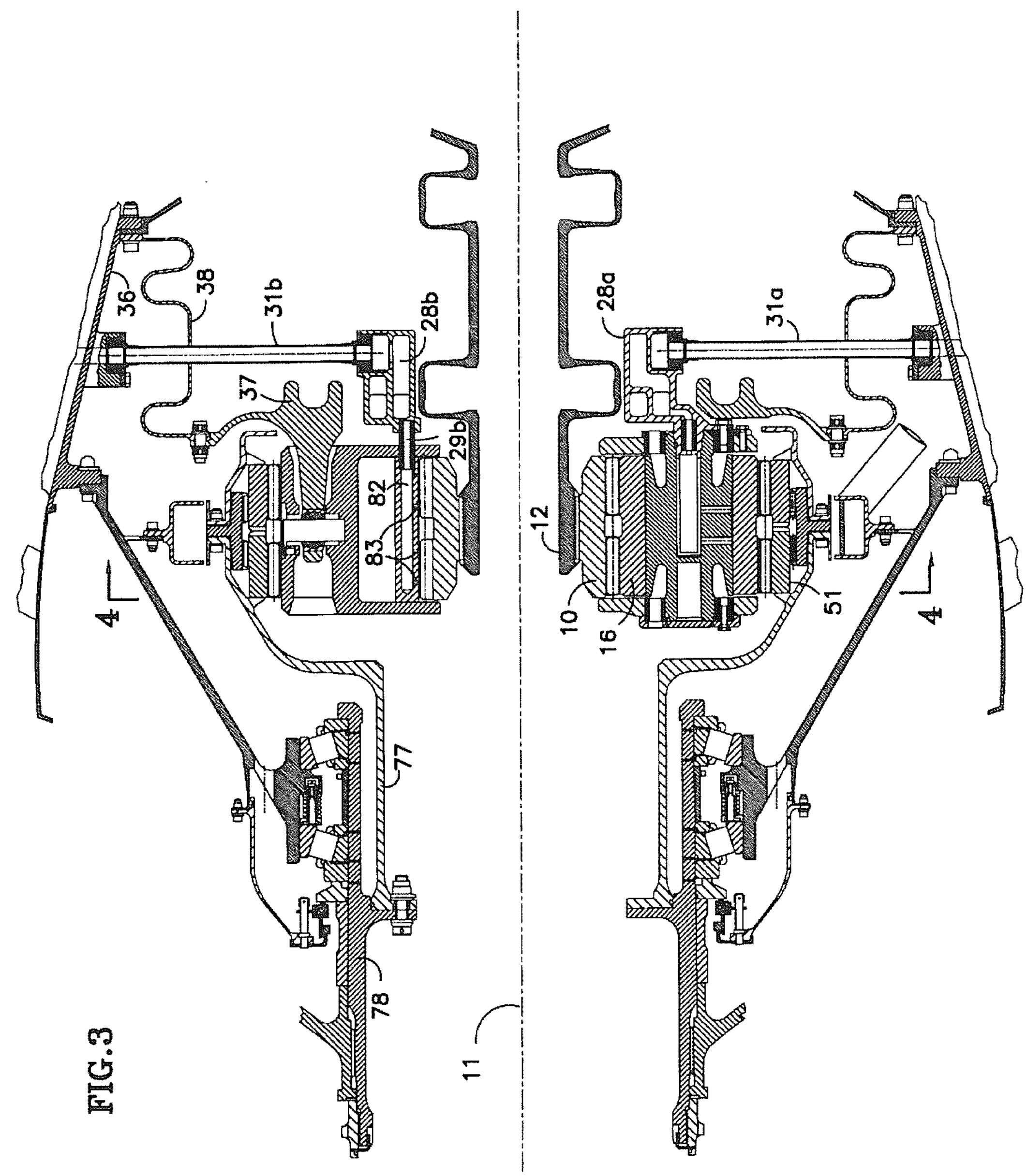
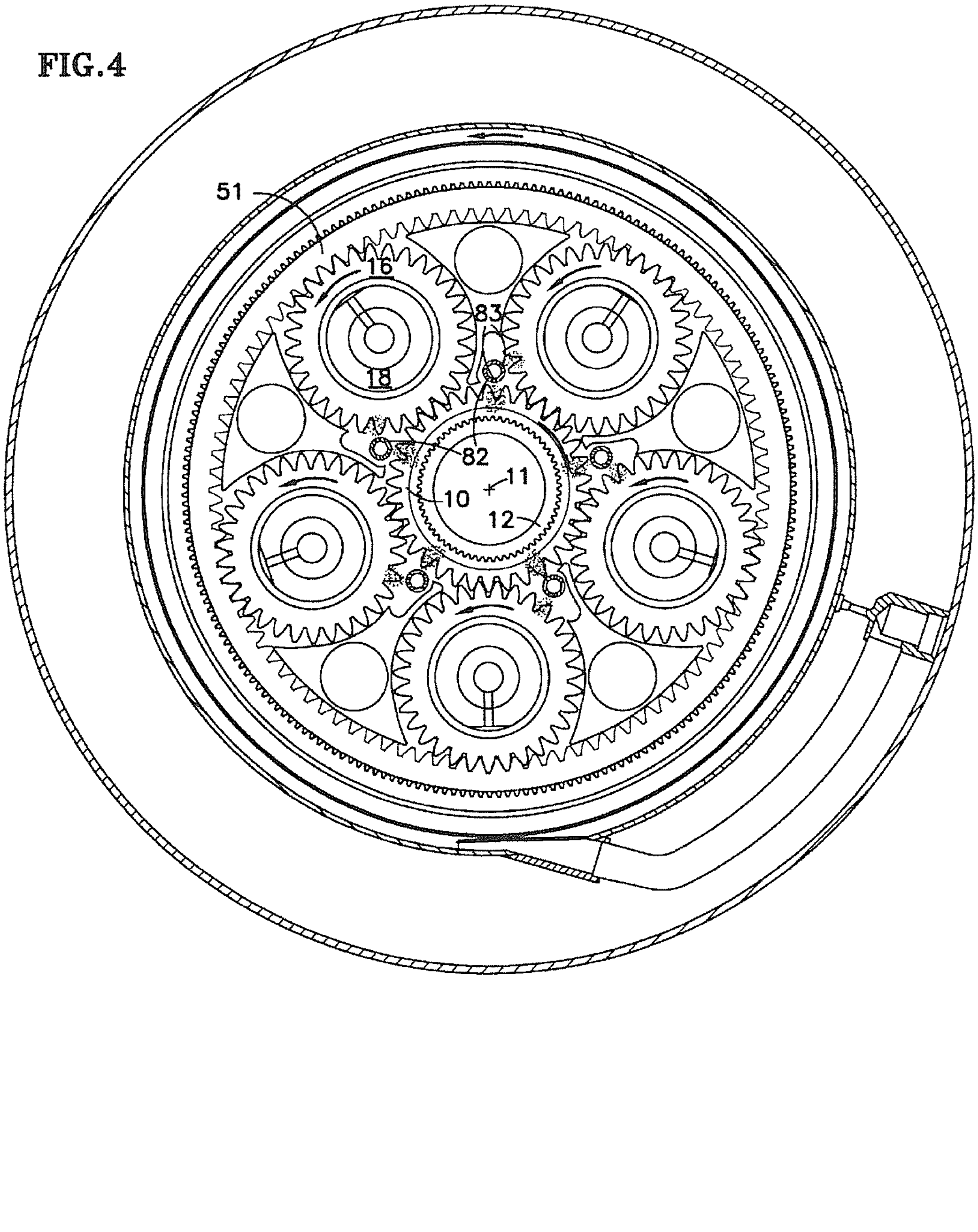


FIG.3

FIG.4



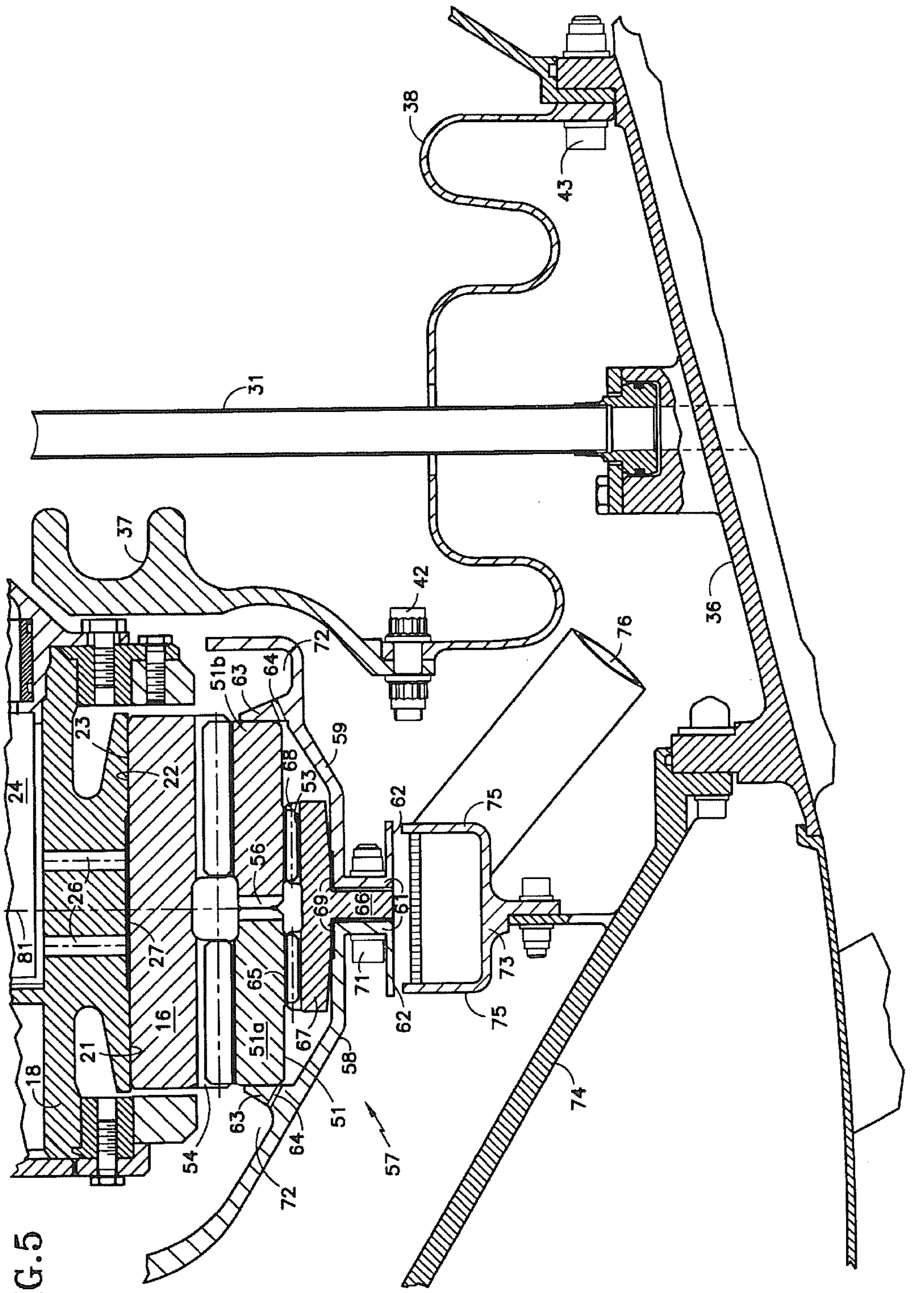

38
43
31
37
42
72
76
36
51b
63
64
23
22
24
59
68
53
62
75
56
26
81
27
69
66
61
71
73
16
65
51a
67
21
18
58
62
75
74
54
63
64
51
57
72

FIG.5

STAR GEAR SYSTEM WITH LUBRICATION CIRCUIT AND LUBRICATION METHOD THEREFOR

TECHNICAL FIELD

The present invention relates to epicyclic gear trains and particularly to an epicyclic gear train configured as a star gear system and having effective, simple means for supplying lubricant to selected components of the gear system and for recovering used lubricant.

BACKGROUND OF THE INVENTION

Epicyclic gear trains are used to reduce rotational speeds in various types of machinery. Depending on the speed reduction ratio desired, an epicyclic gear train can be configured as either a planetary system or a star system. A planetary system includes a central sun gear and a set of planet gears rotatably mounted on a gear carrier by bearings. The planet gears are circumferentially distributed about the periphery of the sun gear so that the planet gears mesh with the sun gear. A mechanically grounded, internally toothed ring gear circumscribes and meshes with the planet gears. Input and output shafts extend from the sun gear and gear carrier respectively. In operation, the input shaft rotatably drives the sun gear, compelling each planet gear to rotate about its own axis and, because the ring gear is mechanically grounded, causing the planet gears to orbit the sun gear. The planet gear orbital motion turns the carrier, and hence the output shaft, in the same direction as the input shaft.

A star system is similar to the above described planetary system except that the gear carrier is mechanically grounded, the ring gear is rotatable and the output shaft extends from the ring gear. Because the carrier is grounded, the "planet" gears cannot orbit the sun and therefore are referred to as star gears. In operation, the input shaft rotatably drives the sun gear, compelling each star gear to rotate about its own axis. The rotary motion of the star gears turns the ring gear, and hence the output shaft, in a direction opposite that of the input shaft.

An epicyclic gear train, whether configured as a planetary system or a star system, also has a lubrication system to lubricate and cool the gear teeth and bearings and to remove used lubricant so that it can be reconditioned (cooled, filtered, de-aerated) and reused. It is desirable to remove the used lubricant as completely and quickly as possible, otherwise the gears continually agitate the residual lubricant. Agitation of the residual lubricant degrades the power transmission efficiency of the gear system and elevates the lubricant temperature, making it more difficult to cool the lubricant to render it suitable for repeated use as a heat transfer medium. If the gear train is a component of an aircraft engine, degraded efficiency is unacceptable because it reduces aircraft range and/or payload. The problem of elevated lubricant temperature can be addressed with larger, higher capacity heat exchangers. However larger heat exchangers are unacceptable because they contribute undesirable weight and consume precious space on board the engine or aircraft.

U.S. Pat. No. 5,472,383 discloses a lubricant supply and recovery system for a planetary gear system. Noteworthy features of the system include a set of lubricant spray bars 32 intermediate each pair of planet gears 10, a set of interplanet baffles 80 each having a trough 82, and a set of collection channels 56. In operation, the spray bars 32 direct lubricant jets 34, 36 toward the sun and planet gears 8, 10. Most of the lubricant 34 passes through the sun/planet mesh. Much of the lubricant that passes through the sun/planet mesh is urged axially outwardly by the gear mesh and directly enters the collection channels 56. The balance of the lubricant that passes through the sun/planet mesh, along with lubricant reflected from the sun gear, is centrifuged into the nearby baffle trough 82, urged through outlets 84 in the planet carrier and finally deposited in the nonrotating collection channels 56. Meanwhile, the planet gears 10 carry lubricant 36 radially outwardly and into the planet/ring mesh. Lubricant expelled from the planet/ring mesh then enters the collection channels 56. Concurrently, pressurized lubricant enters the narrow bearing annulus (unnumbered) defined by the outer surface 44 of each journal bearing 16 and the inner surface 46 of the corresponding planet gear 10. Lubricant discharged from the bearing annuli enters the collection channels 56. Lubricant collected by the channels 56 enters a drain line 62, which conveys the lubricant to the lubrication system coolers, filters and de-aerators.

Despite the merits of the above described planetary lubrication system, it suffers from at least five shortcomings when applied to a star system. First, it relies on the centrifugal forces arising from carrier rotation to evacuate used lubricant. These forces are absent in the star system because the gear carrier is mechanically grounded. Second, the disclosed planetary system suffers from the complexity of two lubricant circuits, one to serve the planet gear journal bearings and one to serve the gear meshes. Both circuits are necessary. The gear lubrication circuit is necessary because the bearing lubricant, upon exiting the bearing annuli, is centrifuged away from the sun/planet mesh and so is unavailable to lubricate that mesh. The bearing lubrication circuit is necessary because the gear lubricant, after having been discharged from the spray bars, cannot be locally repressurized and introduced into the bearing annuli. Although the two circuits may share certain components (e.g. coolers, pumps, filters and de-aerators) other components (e.g. supply lines, spray bars) are unshared and introduce unwelcome complexity. Third, the presence of the dual lubricant circuits dictates that sufficient lubricant be available to concurrently supply both circuits, a distinct disadvantage in aircraft applications where space is at a premium and excess weight is always undesirable. Fourth, because the bearing lubricant cannot serve the gear meshes and vice versa, lubricant must be supplied in parallel to both the bearings and the gear meshes. As a result, a greater quantity of lubricant enters the gear system than would be the case if the bearings and gears were lubricated in series. The excess lubricant can exceed the lubricant evacuation capacity of the gear system thereby increasing lubricant residence time. The increased residence time provides additional opportunity for lubricant agitation and the concomitant loss of transmission efficiency and increased lubricant temperature described above. Fifth, the bearing lubricant and gear lubricant typically originate from a common source. This makes it impractical to customize the lubricant temperature to optimally satisfy the requirements of both the bearing annulus, which requires relatively cool lubricant, and the gear meshes, which benefit from warmer lubricant.

What is needed is a simple lubrication system for supplying lubricant to both the star gear bearings and the gear meshes in a star configured epicyclic gear train and for quickly and effectively evacuating used lubricant.

SUMMARY OF THE INVENTION

According to one aspect of the invention, a star configured epicyclic gear train includes a set of inter-star baffles for constraining the flow of lubricant and a lubricant circuit that

serves as the exclusive means for successively lubricating the star gear bearings, the sun/star gear mesh and the star/ring gear mesh. According to a second aspect of the invention, a star configured epicyclic gear train includes inter-star baffles and two lubricant circuits, a primary circuit and an auxiliary circuit, that share a common lubricant discharge path. Ideally, the auxiliary circuit is selectively operable. The invention also embraces methods for lubricating a star configured epicyclic gear train and for effectively evacuating used lubricant.

One significant advantage of the invention is that it promotes evacuation of used lubricant from a star gear system despite the absence of gear carrier rotation and the accompanying centrifugal forces.

A second significant advantage of the invention, particularly the single circuit variant, is its simplicity.

A third advantage of the invention is that it conserves space and reduces weight by minimizing the quantity of lubricant that must be carried on board an aircraft when the inventive gear system is used as a component of an aircraft engine.

A fourth advantage of the invention is that it minimizes the quantity of lubricant requiring evacuation from the gear system and therefore helps to boost efficiency and attenuate undesirable lubricant temperature rise.

A fifth advantage is that the lubricant temperature gradually rises as the lubricant proceeds through the gear system. The temperature rise conforms to the lubricant temperature requirements and temperature tolerances of the gear system components.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a cross sectional side elevation view of the inventive star gear system with a single lubricant circuit.

FIG. **2** is a view taken in the direction **2**—**2** of FIG. **1**.

FIG. **3** a cross sectional side elevation view of the inventive star gear system with primary and auxiliary lubricant circuits.

FIG. **4** is a view taken in the direction **4**—**4** of FIG. **3**.

FIG. **5** is a view of the gear train of either FIGS. **1** or **3** enlarged to illustrate certain details of a lubricant discharge path.

BEST MODE FOR CARRYING OUT THE INVENTION

Referring to FIGS. **1**, **2** and **5**, a star gear system for reducing shaft speed in an aircraft engine includes a sun gear **10** rotatable about an axis **11** The sun gear is driven by an input shaft **12** coupled to the sun gear by a spline connection **13**. The input shaft includes flexible elements **14** to minimize misalignment between the intermeshing teeth of the sun gear and a set of star gears. The flexible elements **14** are substantially as described in U.S. Pat. No. 5,433,674, the contents of which are incorporated herein by reference.

The gear system includes a plurality of star gears **16** and a nonrotatable star gear carrier **17**. Each star gear is rotatably mounted on the carrier by a journal bearing **18** so that the star gears surround the sun gear and are each engaged with the sun gear to define a sun/star mesh **19**. The bearing **18** is preferably a load equalizing bearing as described in U.S. Pat. No. 5,102,379, the contents of which are incorporated herein by reference.

A narrow annulus bordered by the radially outer surface **21** of each bearing **18** and the radially inner surface of each star gear **16** defines an interface **23** between each bearing and its respective star gear. The radially inner surface of each star gear is coated as described in U.S. Pat. No. 5,685,797, the contents of which are incorporated herein by reference. Axially and radially extending lubricant conduits **24**, **26** penetrate each bearing. The conduits **26** communicate with a shallow, local recess **27** formed in the outer surface of each bearing. Each conduit **24** communicates with an annular lubricant distributor **28** by way of fittings **29**. A supply tube **31** connects the distributor to a source of lubricant, not shown.

The gear carrier **17** includes forward and aft side plates **32**, **33** having circumferentially distributed openings **34**, **35**. The carrier is mechanically grounded to an engine case **36** by a torque frame **37** and a flexible coupling **38**. The torque frame **37** has a set of axially extending fingers **39** that project through the openings **35** in the aft side plate. A spherical bearing **41** couples each finger to the torque frame. The radially outer end of the torque frame is connected to the forward end of the flexible coupling by a set of fasteners **42**. The aft end of the coupling is connected to the engine case by another set of fasteners **43**. to mechanically ground the carrier. The torque frame is substantially as described in U.S. Pat. No. 5,466,198 with the exception that the present torque frame conveys torque reactions from the carrier **17** to the case **36** (by way of the flexible coupling **38**) whereas the referenced torque frame conveys torque and rotary motion from the carrier of a planetary gear system to the gear system output shaft. The contents of U.S. Pat. No. 5,466,198 are incorporated herein by reference. The flexible coupling **38** is substantially as described in U.S. Pat. No. 5,433,674 with the exception that the present coupling conveys torque reactions from the nonrotatable gear carrier **17** to the case **36** whereas the corresponding coupling of the reference patent conveys torque reactions from a nonrotatable ring gear to an engine case.

As seen best in FIG. **2**, the gear carrier also includes a set of baffles, generically designated **44** and individually designated with letters A, B, C, D and E. Each baffle is disposed between two of the star gears. Each baffle has an ascendent flank **46** adjacent the ascending (radially outwardly progressing) side of one of the adjacent star gears and a descendent flank **47** adjacent the descending (radially inwardly progressing) side of the other of the adjacent star gears. The baffle flanks **46**, **47** are each arcuately contoured so that the clearance between each flank and the adjacent gear teeth is as small as reasonably possible. The descendent flank **47** of each baffle is foreshortened to define a roughly triangular, axially extending space **48** bordered by the baffle and by sectors of the sun gear and adjacent star gear. It is theorized that the space **48** may facilitate distribution of lubricant in the axial direction. A depression **49** in each baffle extends axially between the forward and aft side plates of the gear carrier. Each depression is substantially parallel to axis **11**, i.e. the radial distance between the axis **11** and a representative depression is constant along the entire axial length of the depression. Unlike the visually similar baffle troughs seen in U.S. Pat. No. 5,472,383, the depressions do not serve as a means for evacuating lubricant from the gear system because the carrier, being mechanically grounded, fails to centrifuge lubricant into the depressions. Any lubricant that splashes into the three upper troughs, A, B and E, will immediately drain out. Lubricant may accumulate in the lower troughs C and D. However, since the carrier side plates are not penetrated by openings at the axial extremities of the depressions or by other openings dedicated to the removal of lubricant, accumulated lubricant will remain puddled in troughs C and D.

A rotatable, ring gear **51** comprises forward and aft ring gear sections **51***a*, **51***b* having internal gear teeth **52** and external spline teeth **53**. The ring gear **51** circumscribes the star gears and engages each star gear to define a star/ring mesh **54**. Circumferentially distributed lubricant discharge ports **56** penetrate radially through the ring gear.

As seen most clearly in FIG. **5**, a ring gear housing assembly **57** includes forward and aft housing sections **58**, **59**, each having a bolting flange **61**, a splash shield **62** extending axially from each flange, and a gear retainer **63** perforated by circumferentially distributed apertures **64**. The housing assembly also includes an antirotation ring **65**. The antirotation ring comprises a support ring **66** and an integral spline ring **67** having spline teeth **68**. Bolts **71** trap the support ring **66** between the bolting flanges **61** so that the spline teeth **53**, **68** couple the housing assembly to the ring gear ensuring that the housing assembly corotates with the ring gear. A series of circumferentially distributed lubricant discharge slots **69** extend axially between the housing sections **58**, **59** and the spline ring **67** and radially between the support ring **66** and the flanges **61**. The slots **69** are distributed circumferentially intermediate the bolts **71**. The gear retainers **63** trap the ring gear axially within the housing assembly and help to define forward and aft annular lubricant collectors **72**. A nonrotatatable, circumferentially extending gutter **73** is bolted to bearing support **74** so that gutter sidewalls **75** are nearly in contact with the splash shield **62**. A drain pipe **76** extends from the gutter to convey lubricant away from the gear system.

The forward section **58** of the housing assembly **57** includes a cylindrical extension **77** bolted to a power output shaft **78**. Tapered roller bearings **79** support the aft end of the output shaft on the bearing support **74**.

The sun, star and ring gears are bihelical gears whose teeth, if extended to gear centerplane **81**, would form a series of apexes. Depending on the rotational sense of the gears, each apex either leads or trails its constuituent gear teeth. Rotation with the apexes trailing is preferred because it forces lubricant in the gear mesh axially inwardly, toward the centerplane **81** to ensure adequate lubrication across the faces of the gear teeth. Rotational arrows signify the rotational sense of the gears in FIG. **2**.

A lubricant circuit extends through the gear system and serves as the exclusive means for supplying lubricant successively to the bearing interfaces, the sun/star mesh and the star/ring mesh. Pressurized lubricant flows successively through the supply tube **31**, the annular lubricant distributor **28**, the fittings **29**, and the bearing conduits **24**, **26**. The pressurized lubricant is then introduced into each annular bearing interface **23** by way of the bearing recesses **27**. Lubricant introduced into a bearing interface spreads out axially and circumferentially to form a load supporting lubricant film between the bearing outer surface **21** and the star gear inner surface **22**. The lubricant is then discharged from the axial extremities of the bearing interface. Substantially all of the discharged lubricant is directed into the sun/star mesh **19**, partly because of the presence of the nearby descendent baffle flank **47** and partly because of suction created by the rapidly rotating sun and star gears in the space **48**. The directed lubricant cools and lubricates the sun and star gear teeth and then is expelled from the sun/star mesh. The nearby ascendent baffle flank then guides substantially all of the expelled lubricant radially outwardly and into the star/ring mesh **54**. The lubricant is then ejected from the star/ring mesh.

The ejected lubricant is then centrifugally channeled away from the gear train. Since the action of the bihelical gear teeth tends to force lubricant axially inwardly toward the gear centerplane **81**, most of the ejected lubricant flows through the discharge ports **56**. The lubricant then flows axially along the spline teeth **53**, **68**, around the axial extremities of the spline ring **67** and then into the discharge slots **69**. The balance of the lubricant enters the collectors **72**, and then flows through the apertures **64** and the discharge slots **69**. Lubricant that enters slots **69** then flows into the gutter **73**. The splash shield **62** helps to confine the lubricant in the gutter. The lubricant then flows into the drain pipe **76**, which conveys the lubricant to the lubrication system coolers, filters and deaerators.

As the lubricant proceeds through the gear system, its temperature gradually rises in concert with the lubricant temperature requirements and temperature tolerances of the bearing interface, sun/star mesh and star/ring mesh. The lubricant entering the interface is relatively cool, about 200° F. The bearing interfaces **23** require such cool lubricant to support the tremendous reaction forces that the star gears impose on the bearings. By the time the lubricant exits the interface, its temperature has increased to about 240° F, a temperature not incompatible with the sun/star mesh, which can tolerate warmer lubricant without compromising gear durability or appreciably affecting heat transfer. By the time the lubricant exits the sun/star mesh, its temperature has increased to about 280° F. The 280° lubricant may be suboptimally warm for the sun/star mesh. However it is at least tolerable for the star/ring mesh, which is the next component to be lubricated. The star/ring mesh has greater tolerance for warmer lubricant because there is less relative sliding between the star gear teeth and the ring gear teeth than there is between the sun gear teeth and the star gear teeth.

FIGS. **3** and **4** illustrate a variant of the star configured epicyclic gear train just described. The illustrated alternative system includes a primary lubricant circuit and an auxiliary lubricant circuit. The components and operation of the primary circuit are substantially the same as the components and operation of the single lubricant circuit described above. However, in the alternative system, the annular lubricant distributor comprises a primary distributor **28***a* that receives lubricant from supply tube **31***a* and an auxiliary distributor that receives lubricant from auxiliary supply tube **31***b*. Axially extending spray bars **82** having spray orifices **83** are disposed between each pair of star gears. Each spray bar is connected to the auxiliary lubricant distributor **31***b* by a fitting **29***b*. The spray bars are part of the auxiliary lubricant circuit.

In operation, the primary circuit operates substantially the same as the single supply circuit described above. The auxiliary circuit introduces jets auxiliary lubricant into the gear system by way of the spray bar orifices **83**. Substantially all of the auxiliary lubricant and substantially all of the primary lubricant, concurrently enter the sun/star mesh. Once introduced into the gear system, the auxiliary lubricant comingles with the primary lubricant and follows a common lubricant discharge path. The common lubricant discharge path extends through the sun/star mesh and the star/ring mesh and is the exclusive means for evacuating both the primary lubricant and the auxiliary lubricant. The comingled lubricant is then channeled away from the gear system by way of the discharge ports **56**, collectors **72**, apertures **64**, discharge slots **69**, gutter **73**. and drain pipe **76** as described above.

The spray bars introduce some additional complexity into the gear system. Moreover, the auxiliary lubricant supplied by the spray bars may result in the accumulation of residual

lubricant, and the attendant degradation of power transmission efficiency and lubricant temperature rise described above. Nevertheless, the auxiliary lubrication system may be of value when the lubrication and cooling demands of the gear system exceed the capacity of the primary lubricant circuit. Such capacity exceedance may occur temporarily in an aircraft engine during operation at or near peak power. However, operation at peak power usually occurs only during the relatively brief takeoff and climb segments of an aircraft mission. The longer duration cruise and landing segments of the mission are flown at lower power settings where the primary lubrucation system is more than adequate and the auxiliary system is redundant. Accordingly, it is beneficial to make the auxiliary system selectively operable.

Although the invention has been described with reference to a preferred embodiment thereof, those skilled in the art will appreciate that various changes, modifications and adaptations can be made without departing from the invention as set forth in the accompanying claims.

I claim:

1. A rotary gear train, comprising:

a rotatable sun gear;

a nonrotatable star gear carrier;

a plurality of circumferentially distributed star gears each rotatably mounted on the carrier by a bearing having an interface with its respective star gear and each engaged with the sun gear to define a sun/star mesh;

a rotatable ring gear circumscribing the star gears and engaged with each star gear to define a star/ring mesh;

a set of baffles, each baffle disposed between two star gears; and

a lubricant circuit serving as the exclusive means for supplying lubricant successively to the bearing interfaces, the sun/star mesh and the star/ring mesh.

2. The gear train of claim 1 wherein the lubricant circuit includes a lubricant collector corotatable with the ring gear.

3. The gear train of claim 2 wherein the lubricant circuit also includes a nonrotatable gutter circumscribing the collector.

4. The gear train of claim 1 wherein the carrier comprises a pair of side plates unpenetrated by openings dedicated to the removal of lubricant from the gear train.

5. A rotary gear train, comprising:

a rotatable sun gear;

a nonrotatable star gear carrier;

a plurality of circumferentially distributed star gears each rotatably mounted on the carrier by a bearing having an interface with its respective star gear and each engaged with the sun gear to define a sun/star mesh;

a rotatable ring gear circumscribing the star gears and engaged with each star gear to define a star/ring mesh;

a set of baffles, each baffle disposed between two star gears;

a primary lubricant circuit for supplying primary lubricant successively to the bearing interface, the sun/star mesh and the star/ring mesh; and

an auxiliary lubricant circuit for supplying auxiliary lubricant to the sun/star mesh and the star/ring mesh;

wherein the primary and auxiliary lubricant circuits include a common lubricant discharge path that extends through the sun/star mesh, the common path being the exclusive means for evacuating both the primary lubricant and the auxiliary lubricant.

6. The gear train of claim 5 wherein the common lubricant discharge path also extends through the star/ring mesh.

7. The gear train of claim 5 wherein the common path includes a lubricant collector corotatable with the ring gear.

8. The gear train of claim 7 wherein the common path also includes a nonrotatable gutter circumscribing the collector.

9. The gear train of claim 6 wherein the common path includes a lubricant collector corotatable with the ring gear.

10. The gear train of claim 9 wherein the common path also includes a nonrotatable gutter circumscribing the collector.

11. The gear train of claim 5 wherein the auxiliary lubricant circuit comprises a set of spray bars, each spray bar disposed between two star gears.

12. The gear train of claim 5 wherein the auxiliary lubricant circuit is selectively operable.

13. The gear train of claim 5 wherein the carrier comprises a pair of side plates unpenetrated by openings dedicated to the removal of lubricant from the gear train.

14. A method of lubricating a gear train having a sun gear, a plurality of star gears each rotatably mounted on a carrier by a bearing having an interface with its respective star gear and each engaged with the sun gear to define a sun/star mesh, and a ring gear circumscribing the star gears and engaged with each star gear to define a star/ring mesh, the method comprising:

introducing lubricant into the bearing interface;

discharging the introduced lubricant from the interface;

directing substantially all of the discharged lubricant into the sun/star mesh;

expelling substantially all of the directed lubricant from the sun/star mesh;

guiding substantially all of the expelled lubricant into the star/ring mesh;

ejecting substantially all of the guided lubricant from the star/ring mesh; and

channeling the ejected lubricant away from the gear train.

15. A method of lubricating a gear train having a sun gear, a plurality of star gears each rotatably mounted on a carrier by a bearing having an interface with the star gear and each engaged with the sun gear to define a sun/star mesh, and a ring gear circumscribing the star gears and engaged with each star gear to define a star/ring mesh, the method comprising:

introducing a primary lubricant into the bearing interface;

discharging the introduced primary lubricant from the interface;

concurrently directing substantially all of the discharged first lubricant and an auxiliary lubricant into the sun/star mesh;

expelling substantially all of the directed primary and auxiliary lubricant from the sun/star mesh;

guiding substantially all of the expelled primary and auxiliary lubricant into the star/ring mesh;

ejecting substantially all of the guided primary and auxiliary lubricant from the star/ring mesh; and

channeling the ejected primary and auxiliary lubricant away from the gear train.

* * * * *